DOCTOR OF PHILOSOPHY

The design and development of a personal evacuation device

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The design and development of a personal evacuation device

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Declaration

The author would like to clearly state that all the work described herein is his own work unless referenced otherwise. In particular the main elements of the work completed by the author can be summarised as: the completion of the literature review;
the determination of materials; the development of test protocols and performance of tests, the development of new design concepts; the prototyping of designs for testing and evaluation; the design and implementation of test facility for dynamic and static evaluation; conducting a range of tests to further the design concepts; the collection and interpretation of all physical test results and the final viability of the design solutions to meet the design and use requirements.

Stephen Griffiths August 2011

Abstract

In September 2001, a terrorist attack on the World trade centre in New York (USA) provided graphic images, of people trapped and jumping to almost certain death, via news media to a watching world. The aftermath of such an event was to focus not only on the anti terrorism aspect but also the ways that should an event occur again, even if not terrorist related, measures were in place to evacuate buildings by a safe means of exit. In the conferences held after the event attention was drawn to an aspect of height safety and rescue that had previously and also until today received little research.

This thesis outlines the work carried out as a research programme which explores possible design solutions to be applied for developing a personal evacuation device. Focus is on single and multiple evacuation designs in order that occupants can be safely evacuated from heights up to 200 m from high rise buildings on land and oil production platforms offshore. In particular the report deals with the design and development of an evacuation device using controlled descent technology.
A literature review is provided focussing on high profile incidents and accident statistics with respect to evacuation. International Standards and test procedures and how they are applied are also discussed, together with technical and physical trends that are being adopted. The literature review is followed by further background information and discussion of the potential application for evacuation devices. Alternative technologies are considered and evaluated. These include hydraulic and friction braking systems.

In addition the work comprises the development, prototyping and testing of a number of new and different designs. An evaluation and determination of compactness together with sewn lifeline technology is carried out and test rigs are developed in order to validate the designs to the current standards of personal protective equipment.

The study not only considers the design and evaluation of the device but also the lifeline used to connect the mass to the device. The aim is to reduce lifeline volume and provide a compact solution to accommodate descents in excess of 50 m. To achieve this, webbing and rope terminations have been developed with material selection and sewing patterns providing design solutions.

The research concludes by summarising the mechanical function against descent speed for various masses. There is a review of several novel design solutions that meet the international requirements for personal evacuation. The provision of information that furthers knowledge and understanding in descent and evacuation.

KEYWORDS: evacuation, descenders, brakes, rescue, dynamics, testing,escape

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It has not been an easy period and their role has been crucial in providing me the incentives to complete the study.

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Nomenclature

$a, b, c,$	Co ordinates on analysis diagrams
l	Length (m)
m	Mass (kg)
r	Radius (m)
t	Time (s)
t_w	Tape thickness (m)
v	Velocity (m/s)
A_1	Pivot point 1 on four shoe pivot brake
A_2	Pivot point 2 on four shoe pivot brake
G	Centre of mass
L	Leading
N	Angular shoe velocity (rev/s)
P or F	Force acting at the centre of the brake pad (N)
T	Trailing
η	Gear efficiency
θ	Angle of tape (deg)
μ	Coefficient of friction
ω	Angular velocity of the brake shoe (rad/s)

Glossary

ANSI	**A**merican **N**ational **S**tandards **I**nstitute
CSA	**C**anadian **S**tandards **A**ssociation
EN	**E**uropean Standard
HSE	**H**ealth and **S**afety **E**xecutive
PPE	**P**ersonal **P**rotection **E**quipment
RGIT	Robert **G**ordon **I**nstitute of **T**echnology
PrEN	A provisional European standard
SATRA	Shoe and alied trade research association

Definitions

Major injury:	More than three days of work (HSE**)**
Suspension trauma:	Body trauma caused by suspension in a harness
Fatal injuries	Persons, who die within 30 days due to an accident (HSE)

Chapter 1: Background

1.1 Introduction

Due to many factors including increase in population, coupled with limited land mass, architects have moved to design structures with minimal footprint, but maximum use. In turn, this has meant the structures have grown taller, transforming the skyline of cities, towns and villages as well as the industrial landscape. This trend has increased the risks associated with personnel rescue or evacuation from structures in an emergency. At first the solution was one of adding fixed structures either within or external to the main structure, such as fire escapes or stair wells. In the more recent past, over the last few decades, there has been a realisation that alternative options need to be explored and developed to meet the challenges of modern living and structures. This has created a demand for equipment and systems that can either mass evacuate or single evacuate persons to safety, reducing the risk of injury or death. This additionally has resulted in a requirement and need to research methods and types of personal evacuation designs and a method to escape quickly and safely with minimal knowledge or training.

The challenge for engineers is to develop equipment that is not reliant upon external power, is lightweight, easily transported and able to be retrofitted. Working in multi- discipline teams in order to gain the other perspectives on the problem such as the type of body device or harness to prevent injury due to suspension and being able to cope with physical disability.

The starting point came from winch, machine braking and clutch systems linked with mountain climbing techniques. This continues to develop and through research and design the need to provide better insight and data to develop safer equipment.

1.2 Background

Until now, developing a personal descent device from structures meant reliance upon data or experience gathered from research, often not completely relevant to the subject matter.
There is an overwhelming need to obtain more information and data and to present it in such a format that designs can be developed more easily and based on informative data and models.
The dearth of information and configuration models, has resulted in designs which are usually bulky or over engineered, the problem being made worse by the lack of representable test data or the ability to easily test descenders in a laboratory due to a lack of sufficient descent height and other restrictions.

Having approached SATRA (Shoe & Allied Trade Research Association -UK notified body) who are tasked with testing any descender devices to comply with the European standards, they admit that they have no method or equipment to test descenders. As a result, descent tests to the standards for this research were set up and performed with an observer from SATRA being present in order to gain experience on descent devices and how they actually perform.
It is desirable to develop a descent controller that can operate under a variety of conditions easily, where the descent velocity is controlled in order to prevent injury, but would be sufficiently quick so as to take the user away from the structure in a controlled manner without further risks associated with prolonged suspension.
The risks could be due to trauma or external influences such as explosions or fire balls.
Consideration must be given to the fact that the person may be physically disabled or injured and may not be able to offer any assistance in the descent. These factors and others have to be considered in arriving at a solution.
The environment is also relevant, so consideration must be given to the area towards which the person is descending. If it is into water from an offshore structure, then the speed of descent could be greater than a descent onto land. It may also be desirable to stop the person reaching the ultimate landing or entry position if that point presents further danger from obstacles that might injure the person.

The intention is to have a design that is automatic and requires no training so that in the event of a requirement to evacuate quickly the evacuee can simply enter the system and escape or be able to operate it for others, basic set up as shown in fig 1.1. Survival from a height depends upon many factors including the height itself, but also the reason for evacuation. It is, therefore, imperative that options are controlled and that the design has been proven both through theory and practice for good repeatability and whilst in use in a panic situation.

A requirement for functionality is to control the descent speeds, whilst covering as large a range of descent masses as possible over the maximum distance without relying on extrapolation of data caused by its lack of data, inadequate theory or the ability to test the mechanisms over long distances.

The descent speed must be such that when the person reaches the ground level he is not travelling at excessive velocity causing injury might, but not slow enough that the person is exposed to secondary hazards such as suspension trauma, building collapse or other hazards. As the height increases, then the suspension time and that for evacuation become more relevant. Trials conducted by the United States of America military into different harnesses showed that the onset of discomfort and potential risk caused by suspension means that the time suspended must be kept to a minimum, if secondary issues are to be avoided (Brinkley,JW, 1991)

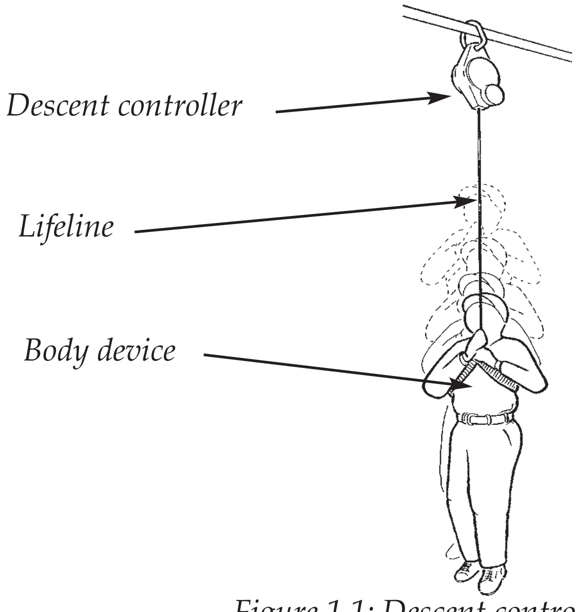

Figure 1.1: Descent controller in use

1.3 Research aims and objectives

The aim of this study is to explore possible design solutions to be applied for developing a personal evacuation device. In this context, a number of brake mechanisms have been modelled together with the development of a suitable lifeline in order to create several scenarios of design to protect the user in the case of an emergency evacuation. This work involves creating test rigs in order to test and develop feasible solutions and to develop theoretical models against which the designs can be evaluated in order to provide engineers in the future with both testing and theory to help in design and development of descent control systems. The research programme was primarily carried out at AYD test facilities in North Wales. AYD is a leading development company in the field of height safety world-wide and the

development of height safety equipment for a number of markets. The work involved the design, prototyping, field testing and laboratory testing of potential technical solutions and can be considered to have the following overall objectives:

1. A major element of a descent control system is the lifeline which must be compact, but must also meet environmental or disaster criteria. Thus, the first objective of the research programme described in this thesis is to develop the lifeline.

2. In developing the lifeline the next issue to be considered is the method of termination in order to again meet the environmental or disaster criteria. Thus, the second objective of the research programme described in this thesis is to develop the termination. The termination specifically relates to the sewing or mechanical provision of attachment to the evacuee and the device, as an example mechanical termination may be achieved by swaging.

3. The prime aim of any descent controller is to control the descent of persons in such a way that they escape in a timely manner, irrespective of their weight or the height of descent. Hence, the third objective is to develop mechanisms that would control the descent speed. Furthermore by prototyping the mechanisms and testing the base line rules, the designs can be investigated and defined.

4. The descent controller has to operate over prolonged periods with constant speed with different masses. Extended tests are required to achieve this objective which is to ascertain how a solution performs in practice. Testing is carried out over different drop distances, using different masses.

5. Having gauged the device performance outside the laboratory, the next step is to see how the designs have performed under laboratory conditions and against appropriate theory. The fifth objective is then to create a test rig and to develop the theory to be able to predict the performance of the designs.

6. The final aim is to test a design using a lifeline with various weights to investigate how changing the gearing between the lifeline and the brake affects the descent speed.

The objective is to see if the design solution can self-regulate its speed such that engineers can predict speed of descent against the gear ratio, thus enabling a larger unit to be constructed which would allow faster initial descent, but would slow the person down as the lifeline is paid out by the device affecting the gearing between the lifeline and the brake.

This is considered of interest in light of the ever increasing heights of structures and the time taken to evacuate from higher levels to the ground level.

1.4 Programme of work

In order to meet the aim and objectives of this project, the following programme of work has been followed:

1.4.1. Initial studies:

This involved a literature search identifying key incidents and International standards.

a. Research into high profile incidents in order to identify major issues in personal evacuation. This was obtained from publications, an internet search and from direct contact with those who operate in the Industry, including in construction and rescue.

b. Research into International standards which are applied to descent control devices and systems. This is obtained from direct participation representing the IMech E on the standards association committee (originally PSM5 now a forum)and direct contact with notified bodies and Standards associations (SATRA,CSA;ANSI).

c. Research and extensive review into products that are available and may soon be available. This was obtained by attending conferences and trade fairs in Europe and North America and by Internet search

1.4.2. Concept and performance phase:

This phase involved the main body of work, leading up to the completion of the research project. In this phase, several approaches were investigated for the lifeline. This included construction, as well as material in order to meet the aims and objectives of the research. Several designs were investigated and samples prepared and tested. The derived data was then studied and used in the development of the ongoing concepts.

The descent control mechanisms were next to be conceptually developed. Following on from lifeline selections, the mechanical functions were considered. Several designs were developed, with prototypes built for testing and evaluation. As with the lifeline, the test data was then analysed in order to develop potential solutions. Many aspects were considered during this phase, including weight, compactness, endurance and repeatability, as well material selection proving useful in the ongoing development of several design solutions.

Test rigs were developed and built to carry out performance tests on a number of functions regarding the lifeline and the mechanical designs. The rigs which were built enabled tests to be carried out under both static and dynamic conditions.

The main body of work can be broken down as follows:

a. An investigation and development of possible solutions for the lifeline was performed. Criteria such as type, size and construction were considered taking into account environmental considerations. Testing was carried out using test rigs developed as part of the study in order to evaluate the lifeline and the effects of different terminations.
Special fixtures were developed to terminate the samples. Prototype testing was carried out on rope, wire rope and webbing lifelines, evaluation of those results was used in the ongoing device development.

b. A study into the various options and methods of creating a controlled descent were performed. This involved making a number of prototypes and testing them on purpose-built test rigs to determine their dynamic performance The data from these test was then used to develop the devices.

c. There were a number of factors considered in the development, including but not limited to the control of descent speed. For example, a user would not be injured by excessive speed or injured by negligible speed (suspension trauma or environmental), the lifeline has to be capable of being deployed safely and capable of withstanding adverse environmental conditions that may occur such as heat or fire.
Material selection and lifeline type played an important role in the development process in order to meet the key aims.

d. Tests were carried out in accordance with EN 341 (BS 341 1993) descent device standard test procedures. A test rig was constructed in order to determine torque, descent speed against drop mass and further developed in order to evaluate the performance of the designs. The test data was compared with theoretical analysis and external testing at a test site, as well as at the purpose-built dynamic test tower.

e. The activation of the descent control design was also considered as part of the development from permanent engagement to motion activation, and from manual control to automatic. Several test pieces were manufactured and tested using the test facilities developed.

f. Tests were also conducted on the concepts at the SATRA Technology centre in Kettering to look at salt spray corrosion tests, overload tests and activation in accordance with EN 341 for several designs.

g. The final objective was to build a prototype and develop a test rig to test the effects of a varying torque on the descent speed for a brake design in a dynamic condition. The data was analysed and compared with theory for inclusion in the design.

There is a dearth of test facilities and capability currently which can inhibit development. There is a firm anticipation that the results and experience gained from this work will facilitate better understanding of test procedures, test methods for descent control mechanisms and lifelines. The results, conclusions and findings can be an available source of information to be used by designers and engineers in the future when developing or considering descent control devices or used by architects in consideration of the escape options in future structural designs.

The next chapter will summarize the literature reviewed in order to guide the requirements and progress the main body of work in this thesis.

Chapter 2 : Literature survey

In the following chapter details will be given of high profile incidents that have occurred in recent times. These are selected to further understand and show the need to carry out research into this important subject. Later, there is an overview of the current International standards and test procedures which are still evolving for this type of development and to assist in understanding of the subject and explain why and how tests are carried out.

Some literature and technical information relating to descent control was found during the various searches carried out and this is discussed. Notified bodies such as SATRA who are tasked with the role of testing to European standards (EN 341) in force since 1992 state that although standards are written there is little capability at present to test to those standards.

2.1 Incident data

2.1.1 High profile Incidents

In 1988, 167 lives were lost out of a total of 229 people on board Piper Alpha platform following a gas explosion on board (Public Inquiry into the Piper Alpha Disaster, Cullen, 1990). The accident took place in only 22 minutes from the initial explosion for the utilities module, including the accommodation block to slip into the sea (figures *2.1* and *2.2* show the before and after condition)

Images removed

Figure 2.1: Piper Alpha before the explosion (gcaptain.2011)

Figure 2.2: Piper Alpha after the explosion (gcaptain.2011)

All routes to lifeboats were blocked by smoke and fire so the decision by the majority was to jump into the sea and hope to be rescued by a boat There were instances of people jumping from the helicopter deck level. It is well documented in courses provided for example by RGIT in Aberdeen, that jumping into water with a life preserver on at heights in excess of 15 m has a high risk of injury to the person (RGIT; 1991). Most were jumping from heights well in excess of 15 m, so decisions had to be made whether to wear a life preserver or not.

In the public Inquiry into the Piper Alpha Disaster, Cullen, The Honourable Lord (1990) made reference to the need to have secondary methods of evacuation, referencing directly descent control devices as a solution for workers in the future.

Even prior to the Piper Alpha disaster, a production platform operating in the Enchova field, Brazil on the 16th August 1984 had a blowout, followed by an explosion and fire.

Two personnel died during the evacuation, 36 killed when the lifeboat lowering mechanism failed and the occupants fell with the lifeboat some 10 - 20 m into the sea. A further 6 workers were killed when they jumped from a height of 30 and 40 m into the sea.

The World Trade Centre Towers 1 and 2 were the subject of a terrorist attack on September 11th 2001 resulting in the death of 2753 people (figure 4). Most high rise structures, i.e above 23 m have built in redundant safety features and mass evacuation of this type of structure is relatively rare so little is known about how quickly one can, or need to evacuate the building. In an explosion or any fire the main method of evacuation are the stair wells or fire escape with all lifts immobilised or automatically returned to ground level.

Levels above 35 m are problematic to reach using conventional fire fighter equipment and techniques such as ladders or extending platform buckets. During the attack on the twin towers (figure 2.3) there were graphic scenes of people jumping to certain death as a last attempt to escape.

Image removed

Figure 2.3: World Trade Centre- 2001 (google September 11.2001)

The pursuit of alternative energy has also brought with it the risk of people being trapped whilst working at the top of wind turbines. With heights up to 100 m and only one means of ingress and egress, the potential problem of people being trapped at the top has meant urgent reviews of the evacuation methods. Descent controllers offer a solution to this potential issue highlighted in figure 2.4.

Image removed

Figure 2.4: Wind generating turbine - 2010 (Martin.2010)

2.1.2 Fall related incidents in the workplace

In the workplace the Health and safety Executive publish statistics relating to incidents at work where injuries or fatalities have been recorded.
Descent controllers, in many instances could help reduce the number of fatalities caused by falling from a height, as well as reducing the number of serious injuries. The problem is identified in figure 2.5 which gives the mortality rate from falling. This does not include any data on mortalities caused by a failure to evacuate and being trapped by structures, fire or explosions.

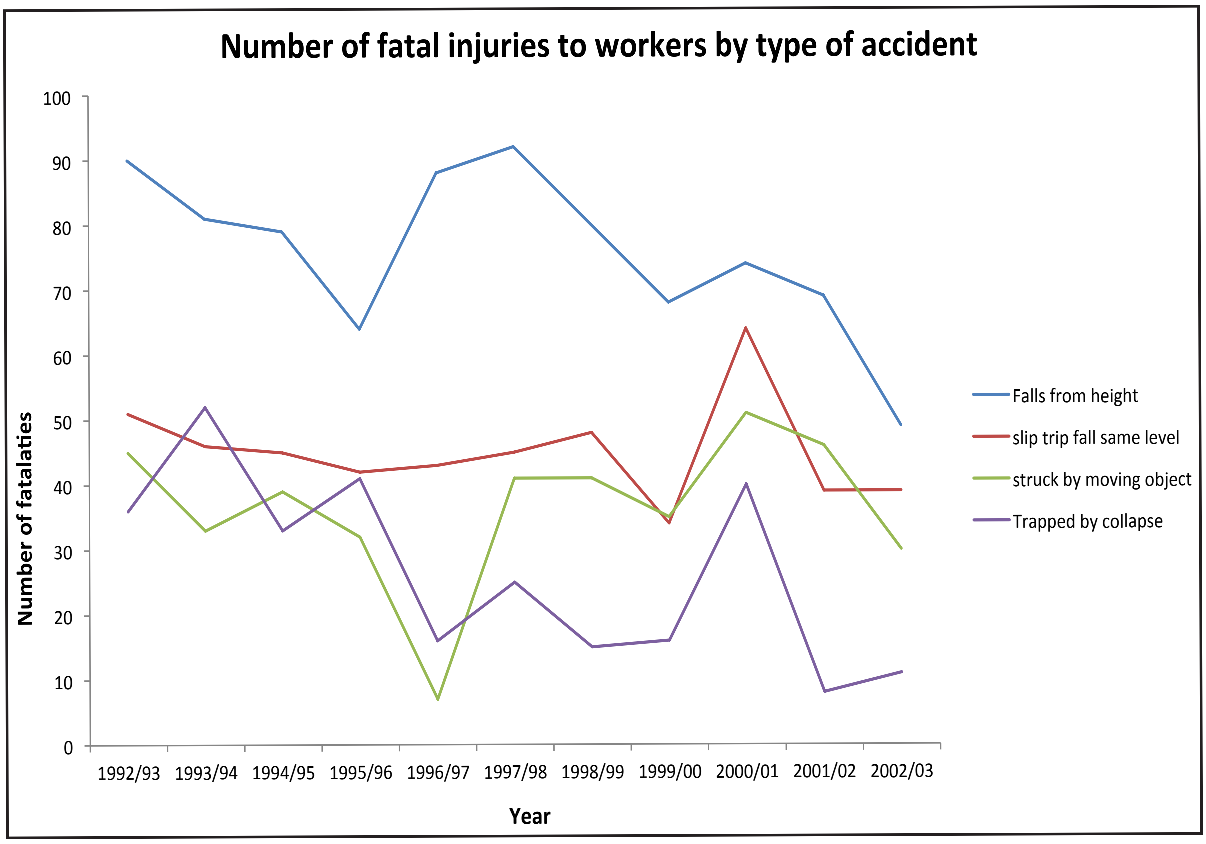

Figure 2.5: Fatal injuries (HSE.1998-2001)

The investigation of the total numbers clarifies the importance of protection at height considering the introduction of the serious injuries and their importance in an even more significant meaning.

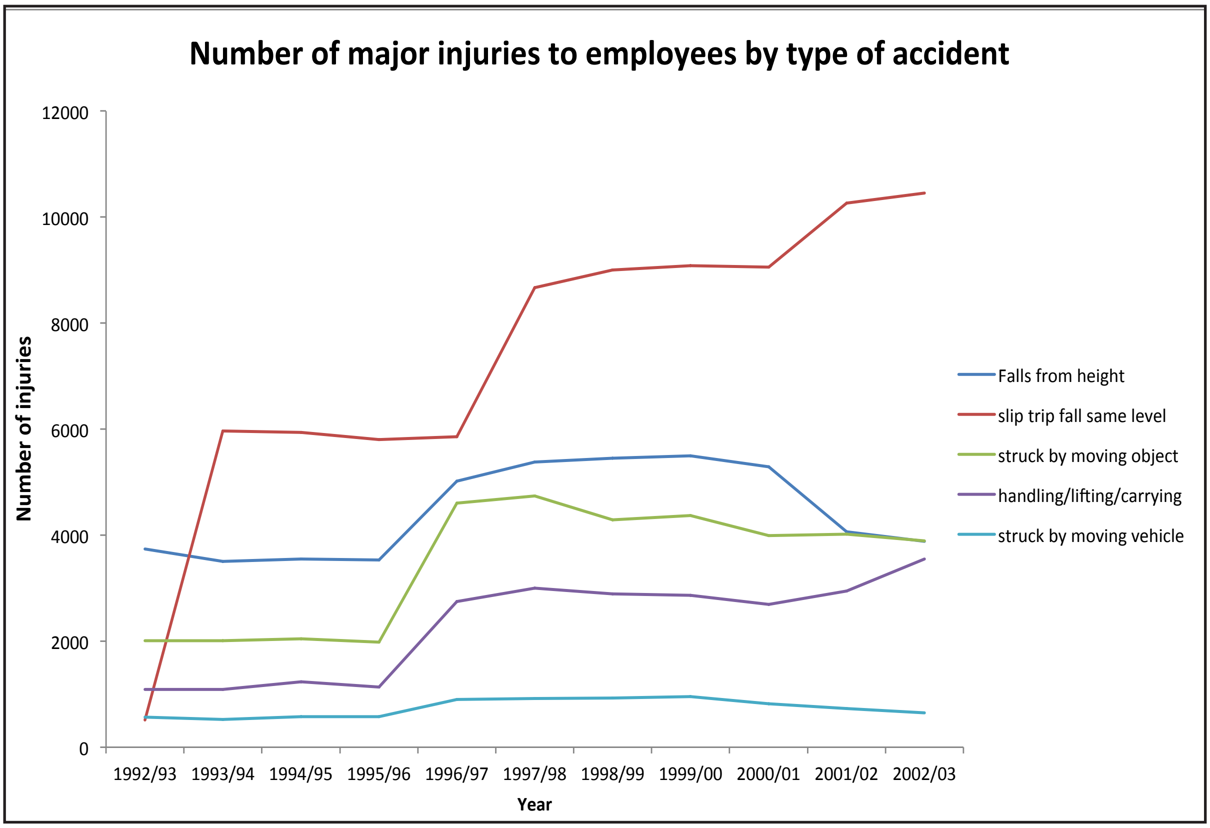

Figure 2.6: Major injuries (HSE.1998-2001)

The trends over the last few decades have tended to be the same according to HSE statistics(figures 2.5 and 2.6). Unfortunately, the data does not identify specifically if the fall from a height was due to an evacuation problem or lack of means of escape. However, recent studies into suspension trauma have highlighted the requirement not to leave persons suspended for any period. This has caused a further need to look at using descent control devices as a means of protection in order that the person is lowered to the ground rather than being left suspended. Risk assessment must now take into account rescue and the use of descenders to fill this role.

2.1.3 Descent control, a current review.

2.1.3.1 The forerunner of descent control device thinking.

In operation today are a number of designs that date back to the original spooled descent design from the 1970's, based on a drum brake, (figure 2.7). It has two large brake shoes that pivot out and based on gearing allow the person to descend, although not generating sufficient friction to stop the descent unless the user is less than 50 kg. The descent also depends upon the amount of rope that has been deployed on the other side, as on longer descents the weight of rope with associated hardware and body harness on the other side of the unit acts as a counterweight and potential snag hazard. The lifeline used is 8 mm diameter and is a polyester sheath over either a galvanised steel or stainless steel wire, which is inflexible and heavy requiring a reel which must be deployed prior to using the device. In the event of a fire it is not certain if the lifeline without the sheath could operate the brake as the brake depends upon the fibres locating within the drum groove.

In appendix 1 the current position is reviewed and expanded further.

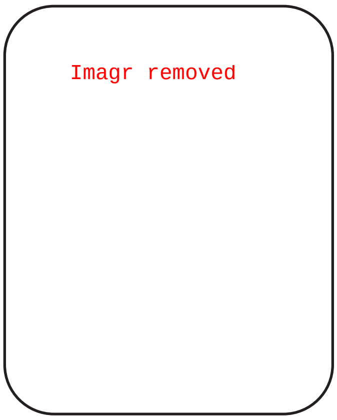

Figure 2.7: High rise escape systems (Barrow Hepburn.(n.d.)

2.1.3.2 Evacuation and prior art search

Prior to an evacuee being able to escape, using a descent controller, a safe point has to be found or negotiated too, several studies into crowd control, and how best to move people in the event of an incident have been carried out.
One such study is known as PEAR (Personal Evacuation And Rescue system) which was developed by three members of the department of computer science, Natl. Chiao Tung University of Taiwan, namely Chu-Yi (Juyi) Lin; Yu- Chee Tseng and Chih-Wei Yi in 2011.
In this proposal satellite Navigation is used in conjunction with mobile devices, television, electronic signage and other devices to instruct persons on the best evacuation route through structures or other environments. By using this technology then it has the ability to constantly update the evacuee with the latest information and provide safe route options.

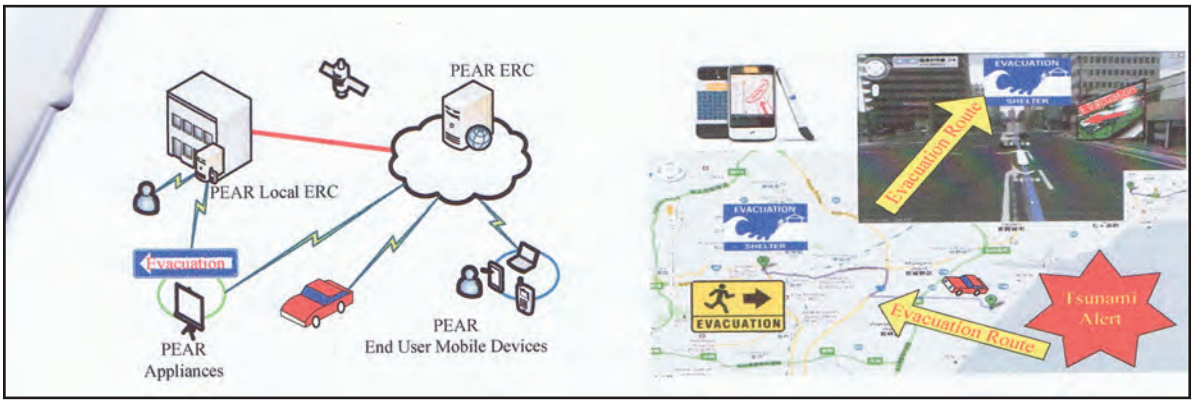

Figure 2.8: The PEAR system graphic
(Natl. Chiao Tung University,Taiwan. 2011)

This type of system has merit when considering land based incidents but may have more problems if implemented on offshore structures such as Oil or gas production platforms. It may also present a way of tracking evacuees and providing instruction directly to them in how to use descent devices, other than that its relevance to this study is limited.
In a paper by Alois Ferscha and Kashif Zia (Lifebelt: Silent Directional Guidance for crowd Evacuation, 2009), they propose the use of a lifebelt for controlling evacuees in a panic situation. The belt would vibrate to commands from the global evacuation control unit, in this way the wearer is directed to the most effective escape route, it was also proposed that the belts would gather data on movement for continual updating and route prediction, this would in theory enable bottlenecks to be avoided. Once more this system is designed to allow the evacuee to navigate to the safest escape point, if at height then that is where the descent controllers would be located (figure 2.9 and 2.10).

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Figure 2.9 Vibrotactile lifebelt (Institute of Pervasive Computing, Linz, Austria : 2009)

Figure 2.10: Microcontroller, Vibra element array (Institute of Pervasive Computing, Linz, Austria : 2009)

In this section consideration is given to the problem of moving people to a safe area in order for them to evacuate safely. With regard to high rise structures there are two principal ways to move from one level to another, the first is ia the stair wells which are located through structures and the other being the Elevators. In the event of an incident the elevators are in general offlimits as a means of evacuation, often programmed to return to ground if a fire occurs. Elevator shafts are a possible evacuation point for the use of descent control devices as an alternative to running down the outside of the structure.

An evacuation assistance device for elevators has been patented in the USA (US 7,963,372 :2011), this relates to a device that controls the elevators and manages their operation in an emergency. The intention is to use the lifts to move evacuees to designated floors for safe exit, this can co-exist with the provision of descenders on these floors for evacuating the building. The device can assimilate information and perform live updates which it can use to move the elevators and people. by channelling people in such a fashion it is hoped that an effective orderly evacuation can be carried out. The device runs alongside emergency broadcasting devices to inform people of the procedures and way out. There is no mention of being able to operate elevators and doors such that the elevator shafts could be used but it would appear that this is an possible extension that could offer alternative options in evacuation if used with descender devices. Figure 2.11 shows the general schematic for the device.

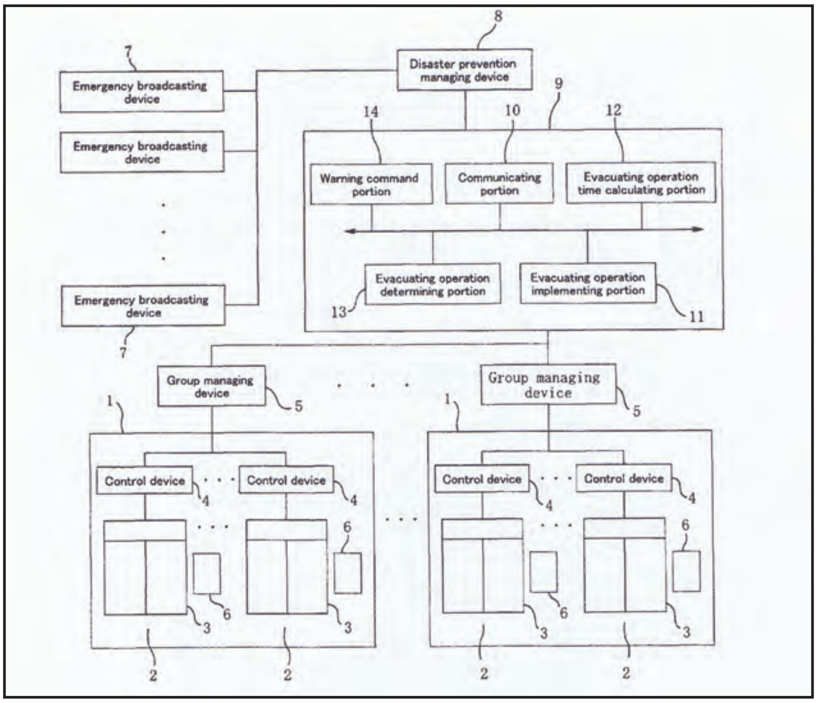


Figure 2.11 Control schematic
Evacuation Assistance Device for Elevator
(US 7,963,372 : 2011)

Moving from the concepts of how to move people to a safe point for evacuation, consideration is now given to the actual methods of descending, in 2.1.3.1 the forerunner of modern descenders is discussed as are several types in appendix 1. In the next part the discussion looks at several devices obtained by the search.
A number of winch type devices have been modified in order to lower a person from an elevated position to a lower level. one such winch was patented in the USA (US 4,493,396 : 1985) figure 2.12.

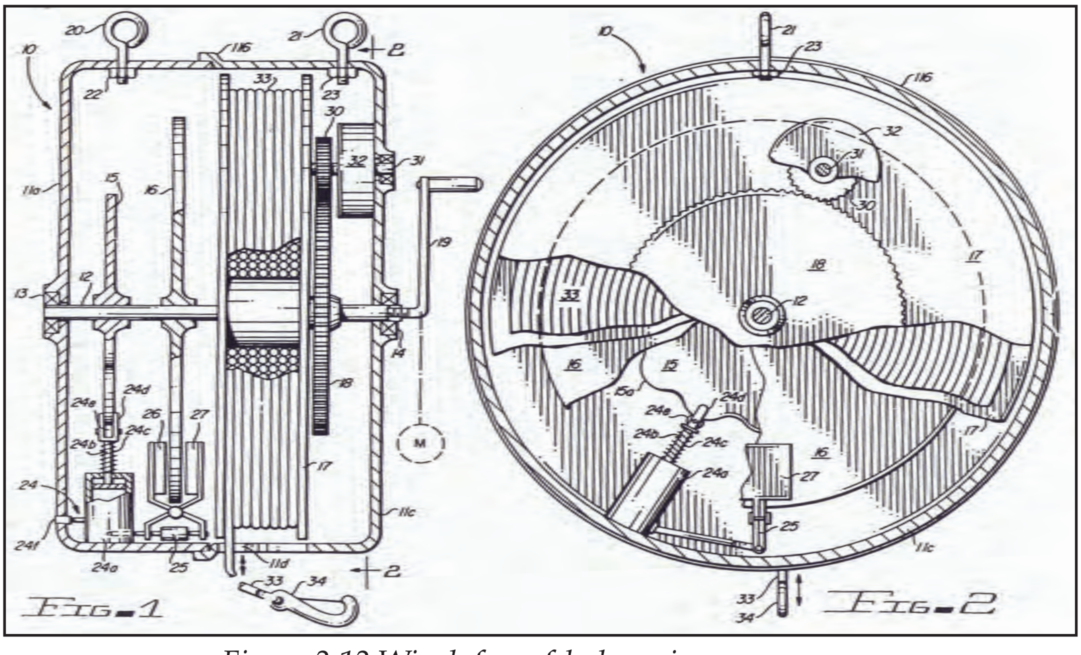


Figure 2.12 Winch for safely lowering a person
(US 4,493,396 : 1985)

The winch in this case has an overspeed brake, in the left hand side to stop the unit if the handle is released. If the brake were set such, that the air piston only applied limited pressure to the disc brake then a degree of descent unaided may be possible.This is not the prime aim though, which is for a rescuer to lower the person and the unit using wire as the lifeline would be very heavy in any event. The brake would be hard to control other than for complete overspeed stop and the unit is not considered practical for automatic descending.
Also based on the winch type of descender is a fire escape apparatus which was filed in the UK by ta-Tan liou and Shiou-Huey Chang (GB 2 238 720 : 1991). The device is

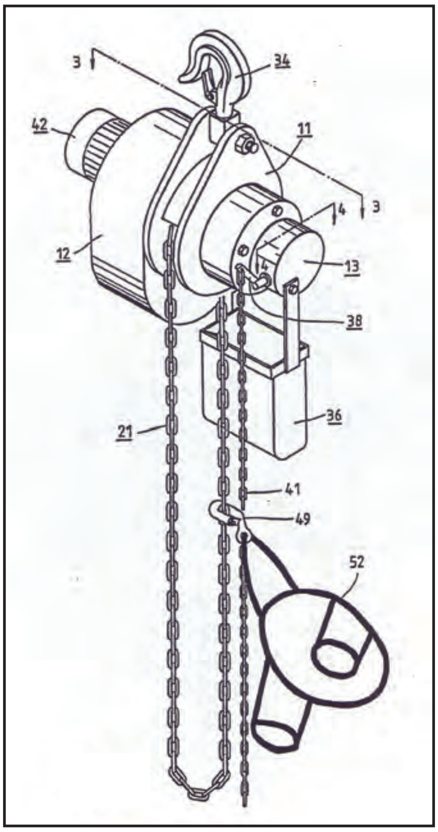


Figure 2.13 Fire escape apparatus (GB 2 238 720 : 1991)

based on the chain hoist, employs a chain, which it is claimed evacuees can clip into the links of the chain, descend either by themselves or multiples. Chain blocks would require a lot of chain which is hard to manage and the feeding is often with problems. It is difficult to determine how the unit would perform in reality, there exists a very complex set of gears with a speed limiting device that is small and appears none compensating. The principal of simply connecting to a link is liked but having studied the arrangements one tends to believe that its use would be an issue if it in fact works at all as a descender. It is an interesting variation on the chain hoist which in the norm operates with ratchets. Figure 2.13 shows the general arrangement.

Portability is a key factor with descender devices, as the unit may need to be relocated in an emergency then compactness and weight are to be monitored. A portable slow descender was patented by takeshi Kikuchi (US 4,986,390 : 1991) figure 2.14.

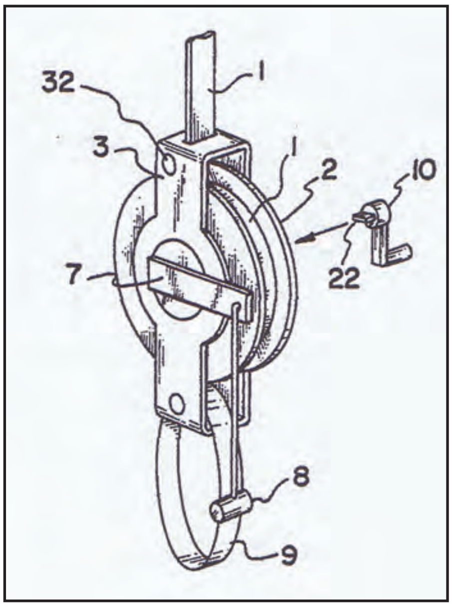


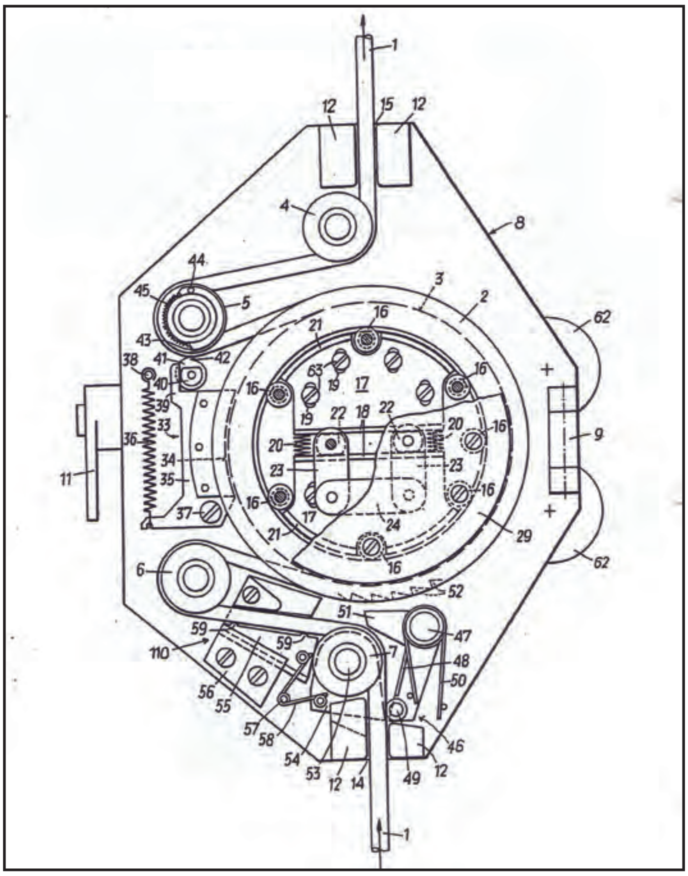
Figure 2.14 Portable slow descender

(US 4,986,390 : 1991)

This invention utilises a drum, on which a webbing is wound the descent is controlled by a lever that applies a brake to the inside of the drum. The person is connected to one end of the device via a strop . No indication is given to the method of lifeline termination at either end or the size of brake, from looking at the sectionals it is hard to see how the device would or could regulate for heat. As the device travels with the person, on descent then temperature and rotating parts may be an issue. The principal is similar to the mountaineering, figure of eight and would require levels of training and device know how to operate it safely. The other issue is the brake regulates the descent but in a panic situation, what would have perhaps been better is a dead man's handle so that you can stop-go stop otherwise you could hang up. The use of tape was however, interesting as most other searches revealed rope or wire as the preferred choice.

Roping type device form the main areas uncovered from the search. One such design was patented in the UK by Hans Bloder (GB 2 057 871 : 1980).

The device relies upon a rope being threaded through the unit and the action of this producing controlled descent due to friction. The unit can be further regulated manually via cams in the device. In the event of rope being fully used or a problem with the unit then a rope sensor in the device would apply a clamp stopping any further rope movement or descent. Rope camming devices put both twist and loading on the rope which can cause premature wear, also as with other roping devices the rope and way it is mounted are critical to the operation. Again training is an issue as is the ability to have multiple descents quickly and easily, figure 2.15.



Roping down device
Hans Bloder
(GB 2 238 720 : 1991)

Rope wrapped around or threaded through a device remains the main part of any search carried out. W. E. Forrest patented Personal high rise evacuation apparatus (US 4,550,801 : 1985. Agin this invention is reliant upon friction of the rope as it passes through the device. Forrest talks of a simple unit for the none trained with in out only in other words the rope is pre loaded and ready to be used once it has been deployed from the bag. Forrest then goes on to cite a second invention which is for the rescue or trained persons. the rescue unit would have the ability for re threading the rope on site and adjustment. At the time Forrest also cites the use of “kevlar’ ropes 5 mm in diameter as an option for his device, this choice would be questionable knowing the poor bend quality of “kevlar”.
Speed is determined by how many wraps there is on the drum, suggesting 3 or 4 would cover most cases, reference to speed per wrap as against weight is not given.

A common theme with rope devices is the cylindrical approach comprising a central stem or drum upon which the rope is wound. Lewis H. Himmelrich patented a device , namely; Descent with a manually operable brake (US 4,474,262 : 1984) figure 2.16, which has a rope around a centre stem, the rope enters at the bottom and exits at the top. the device has a brake lever which is operated manually by the evacuee, allowing additional control over the descent speed, the lever simply clamps the rope at the bottom. If clamped hard then the descent would stop but as with the previous device reviewed if used in a panic situation, then the descent may stop completely unintentionally with inherent problems of how to get the user to release and lower.

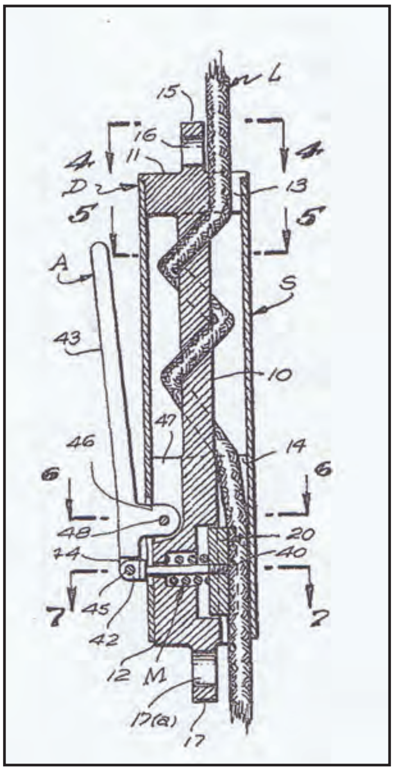


Figure 2.16 Descent with a manually operable brake *(US 4,474,262 : 1984)*

Further information on types of descenders is given in Appendix 1in particular A1 8.3 to A1 8.5 inclusive.

A further variation on the descender with manual override, or more accurately in a stop-go-stop configuration is given by US patent, Descent control device with deadman brake (US 4,883,146 : 1989).

The device was invented by Horace M. Varner and Ernest L. Stech and differs from the previous device as the user is required to move an outer cylinder downwards to effect descent, if one lets go then the descent is stopped, if one pulls it to far again the descent is stoppled. this type of device has what is described as a dead man brake, (figure 2.17 and appendix A1.8.4).

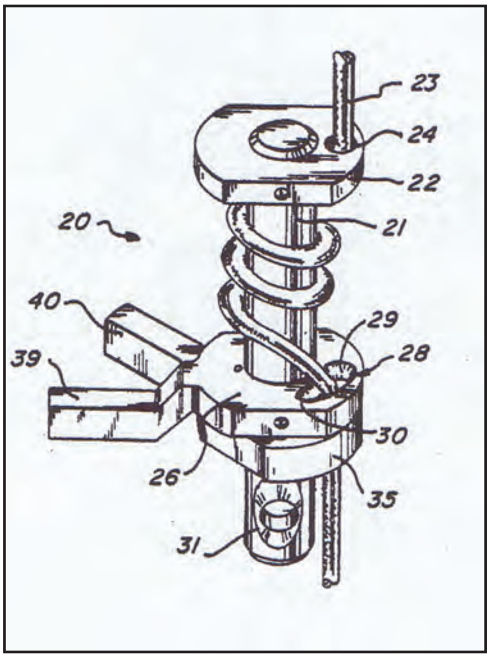


Figure 2.17 Descent Control device with deadman brake *(US 4,883,146 : 1989)*

Again the rope is wound around the centre stem entering at the bottom and exiting at the top. The number of turns in line with other devices of this type dictates the descent speed and range. One problem with this type of device is the issue of dead man brake, which is designed to make the unit more easy to use, however, if the user were to collapse or encounter any problems then they would be left suspended which in itself is a problem and could require a full scale rescue itself.

A deviation on the wire on the drum theme is contained in US Patent 6,672,428 :2004 by Boris Gelman.This device is attached to the back of the evacuee and as they descend the wire is payed out, the drum rotates and the speed control is obtained by moving a hydraulic fluid through a channel. the hydraulic fluid is moved due to an oil damper pumping the fluid this pumping action acts as a retarding force causing the slowing of the evacuee. Questions arise as to the effect of temperature rise in the unit and the true action of the damper which provides the resistance. Also it would be of interest to know the descent speed achieved for various masses and the descent height achievable. With all the wire being paid off from one end of the spindle, this may put a twist into the user as one descends.

Figure 2.18 shows the general arrangement of the hydraulic device, from which one can see that the person is connected via a yoke connected to part 42, giving operational concerns due to twist.The hydraulics are also not direct driven as it relies on a pump which may be a further risk as no back up exists.

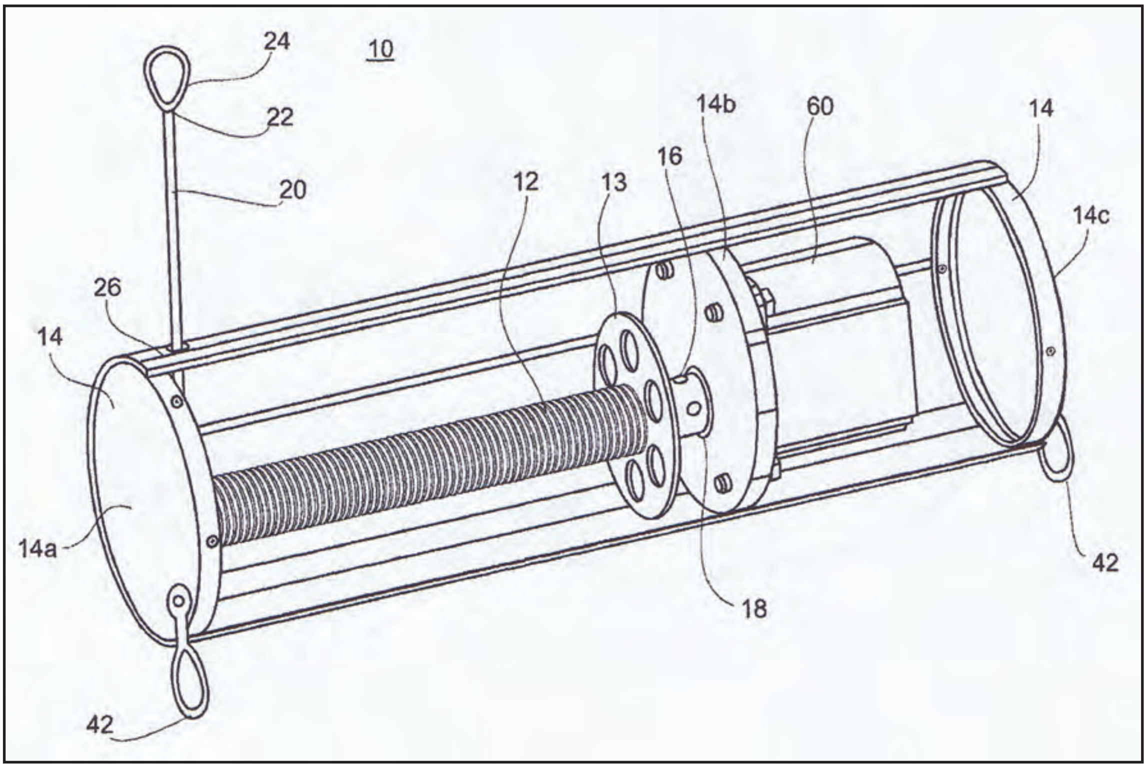


Figure 2.18 : Personal descent apparatus
(US 6,672,428 : 2004)

Evacuation from height can take other forms, in recent years a variety of slides and constriction tubes have been tested. The slide is well established and familiar to most as used on commercial aircraft for many years. In the USA a patent was filed by Alexandre Targirff and Dean H. Staudt for an inflatable evacuation slide with adjustable decelerator (US 6,298,970 : 2001) figure 2.19.

Where this differs from other slides is the inclusion of decelerator ribs which are designed to slow an evacuee down. The more ribs used the slower the descent, it is unclear at what point does the inclusion of additional ribs actually inhibit the descent such that multiple evacuees collide. also the slide would by its own construction have height limitations and comes with stowage and deployment issues. Portability would appear not possible as would the activator required to deploy the slide, if based on liferafts then compressed gas adds weight and bulk further hindering the use in an emergency from none fixed locations. The materials of construction may also be a concern in the event of fire and heat.

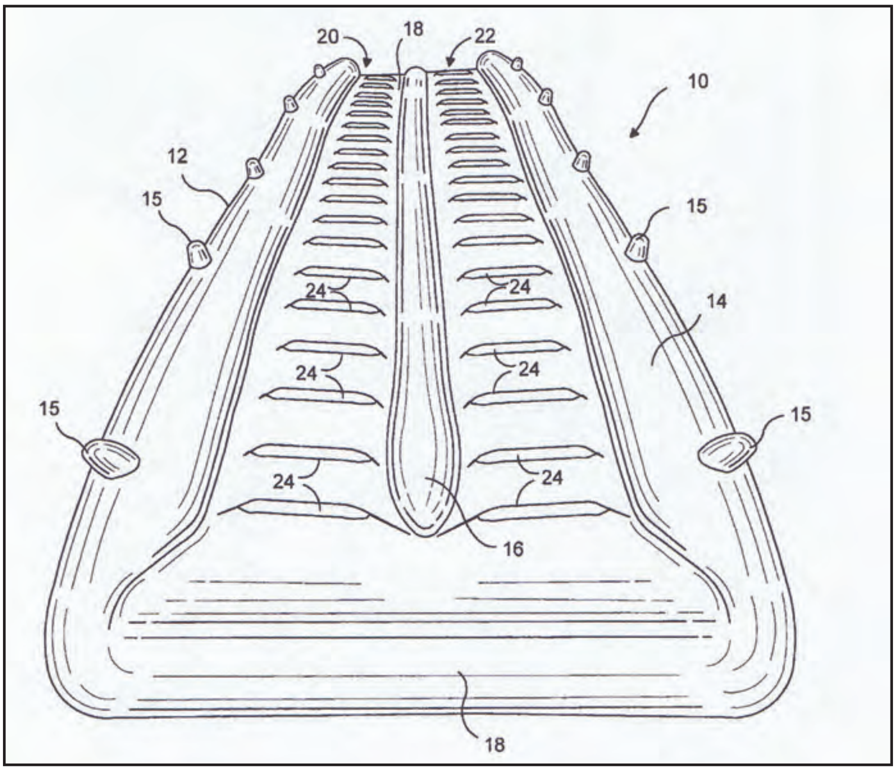


Figure 2.19 : Inflatable evacuation slide with adjustable decelerator
(US 6.298,970 : 2001)

During the search a US patent was filed by Linda Jones Hunt for an emergency evacuation unit (US2010/0065690 : 2010). Figure 2.20 shows the general arrangement for the unit.

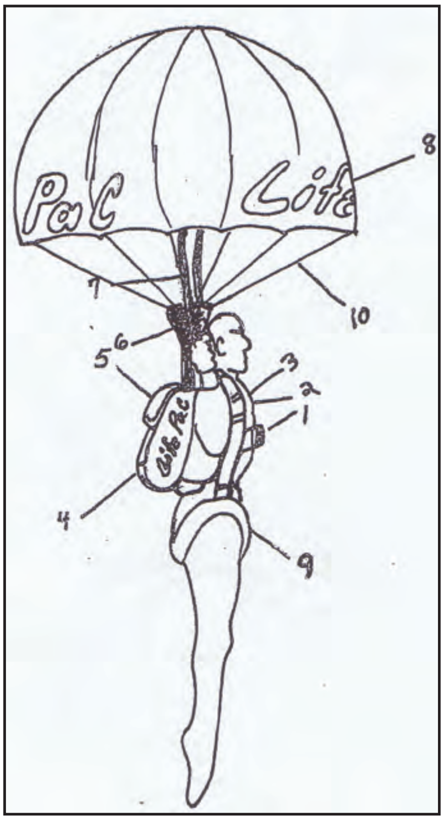


Figure 2.20 : Emergency evacuation unit
(US 2010/0065690 : 2010)

This device is an easy to deploy parachute which incorporates sensors which will deploy the parachute after a time period, or can be deployed manually. The unit is light and compact but would suffer from thermals and updraughts if used near tall buildings, also for an untrained user the concern would be, how does one steer the chute such that one does not drift into a problem area such as a fire. It is also unclear if the evacuees would have the confidence to strap the chute on and jump.

2.2 International standards search

2.2.1 European - EN 341 :1992 (BSI.1992)

The EN standard's main aspect is the energy rating or classification for the descent device. It splits into four categories A,B,C and D where A is the highest rating for continuous use and D is for single use only. It also relates the classification to the static strength requiring far higher for Category A (12 kN) then for category D it is far lower (5 kN).

The aim is to try, as far as is reasonably possible, to have a distinction between one- off personal devices and multiple use devices.

The standard forms the base for all other standards and is considered to be the standard that should be first applied to all research.

The problem is that the standard contains limits and test protocols that are in practice very difficult to achieve as even notified bodies in the various European Community Countries have no way of testing the units. In order to carry out this research, several purpose designed test rigs have had to be constructed in order to verify the design concepts. The descent velocity aim is 2.0 m/s and this has to be achieved over a weight range from 30 kg to 150 kg.
Energy tests are carried out with a 75 kg weight, based on a 100 m drop and the test would have to be repeated 7 times to achieve 0.5 E6 J Class C rating, whereas the test would be run over 20 m using a 100 kg mass to achieve 0.02 E6 J class D rating. The energy rating requirement highlights the substantial difference in testing the devices to achieve each class.. In addition Class A - C require a static strength of 12 kN where as the Class D device only requires a static strength of 5 kN - Table 2.1 provides an overview of the standards.

2.2.2 Canadian - CSA Z259.3-99 (CSA.1999)

As with the European standard, the Canadian standard is based on energy but class D in the European has no equivalent in this standard. As would be expected the additional tests relate to low temperature due to the Canadian environment with tests carried out after conditioning to - 30 ^{0}C (table 2.1).

2.2.3 American - ANSI Z359.4-2007 - (ANSI.2007)

In this instance the standard has only two categories single use and multi use. Upon reviewing the standard it is based on a combination of the other standards which, as with the Canadian standard, has a test protocol that is based on heavier weights (300 lbs or 136 kg) than the European standard (100 kg).The ANSI standard is the latest attempt by International bodies to try to regulate what descent equipment is being used and is a reflection of the growing need for evacuation devices. The American engineers following on from the Twin Tower attack in 2001, are looking more and more at evacuation and rescue, (table 2.1)

	EN 341: 1992	ANSI Z359.4-2007	CSA Z259.3-99
Class / Type:	A, B, C & D	Single Use & Multi Use	1E, 2E, 2W & 3W
Material Requirement:			
- Strength of Wire rope	1770 N/mm^2	3000 lbs (13.3 kN)	13.4 kN
- Strength of Strap		3000 lbs (13.3 kN)	13.4 kN
- Case Displacement of	<15 mm over 2 m strap length (test accord-		
- Elongation of Strap	<8% (test according to 5.2 of EN341:1992)		
Descend Line Termination	12 kN forClass C		4kN for 3 minutes at the tail end of descend
	5kN for Class D		
Holding Load of Hand Oper-	Max 120 N to hold 80 kg test weight		
Static Strength	Maintains 12 kN (5 kN for Class D) for 3	Maintains 2700 lbs (12 kN) for 1 minute	Maintains 12 kN for 10 minutes
	Maintain 5kN for 3 Minutes Perform test be-		
Descent Energy Tests	Class C: (0.5E6 J) 75kg test weight, 100 m	Multi Use: (300,000 ft-lbf) 310 lbs test	136kg test weight, Descend height = device
	Class C: (0.5E6 J) 75kg test weight, 15 m	Multi Use: (300,000 ft-lbf) 310 lbs test	136kg test weight, Descend height = device
	Class D: (0.02E6 J) 100kg test weight, 20	Single Use: (30,000 ft-lbf) 310 lbs test	136kg test weight, Descend height = device
			Extra tests for wet, cold & wet/cold conditions
Functional Test (with condi-	After Energy Test, this test shall be carried	After Energy Test, this test shall be carried	
	Class C: 30 kg & 150 kg test weight, De-	130 lbs & 310 lbs test weight, Descend	
	Class C: 30 kg & 150 kg test weight, De-	130 lbs & 310 lbs test weight, Descend	
	Class D: 30 kg & 100 kg test weight, De-	130 lbs & 310 lbs test weight, Descend	
	Meet Descend Velocity below	Meet Descend Velocity below	
	Extra test for wet condition required for	Extra test for wet condition required	
Descent Velocity	Class C: 0.5 m/s - 2.0 m/s.	1.6 fps - 6.6 fps (0.5 m/s - 2.0 m/s)	Type 1E: <2.0 m/s over solid ground, <4.0 m/s
	Class D = to or less than 2 m/s		
Temperature Rise of De-	<48°C any parts touched during energy test		<65°C any parts touched during energy test
Manual Descender (Hand	<2.0 m/s after control device of hand oper-	<2.0 m/s after control device of hand oper-	
Special requirement for	Shall be designed in a way that it cannot be		
Dynamic Strength		220 lbs (100 kg) test weight, 2 ft (0.6 m)	220 lbs (100 kg) test weight, 2 ft (0.6 m) free
Salt Spray			No evidence of corrosion per ASTM B117
Resistance to Slippage			Type 2E & 2W only
			Hands-Free Lockoff: 3 kN applied to device
			Panic Lockoff: 3 kN applied to device (with 45
Residual Static Strength			The descend line shall retain a min 90% of its

Table 2.1 - International standards relating to descent control

2.3 Summary and comment

There is a dearth of information relating to descent control devices in the public domain which could be drawn upon, much of the information that was found unfortunately had little relevance to the main aims of this research. High profile disasters are recorded, as are the various groups set up to try to improve the situation for the future. However, the specific information is either restricted as commercial, in confidence or simply does not exist. From discussions with several bodies and notified bodies (authorised persons man dated under European legislation to test safety products to ratified standards) world- wide it is clear that little research has been done and there is a lack of data or test facilities (Appendix 6). A review of proposed changes to the European standard has also been considered as a PREN (pre ratified standard) publication,the main points remain. However the way designs are to be classified between evacuation and rescue is the main change as designs now stated for rescue under proposed changes would not require EN testing.

The move is to encourage new designs that do not or can not fit within the original EN 341 scope, but can perform rescue tasks.

The testing regime for descenders remains but in all discussions and from all searches the ability to test to the standards remains a problem.

Chapter 3 :The design approach for a personal evacuation device

3.1 - Introduction to principles

Research into the area of descent control as used for evacuation and rescue can be an effective way to avoid injury. Improvements in descender design in particular for use by untrained personnel has been very limited to date. Some examples are cross over designs from mountaineering, such as figure of eight descenders and belay devices, figure 3.1 shows such a device with origins in mountaineering, although serving a need are not considered adequate for mass evacuation or the unskilled user.

The approach including some passive improvements has been held back due to other considerations such as cost and the lack of test facilities to try out and develop designs. The descent control unit requires either movement or manual control in order to operate.

Due to the nature of where they would be used, the use of an external power source is not very practical or desirable. The fact that the environment can change considerably even during use is a factor that influences design.

The design has to be made in such a way that even if not functioning perfectly, it would still be able to allow a user to descend safely. If the units and their operation have no back up, so the risk of injury or worse would always be prevalent and the designer has to be aware of this shortcoming.

The proposed research designs described in this thesis are based on reliable activation and use mainly gravity and centrifugal forces to effect the descent. The designs are able to be reused without resetting, which eliminates the use of manual operated devices as in the event of trauma the device has to still function.

Image removed

figure 3.1 : A manual descender developed during the course of this research

3.2 Evacuation

The main use for any descent device is evacuation either as a result of an accident, suspension or escape. The current research looked at simple manual devices, semi-automatic devices and fully automatic devices for use over land and water (figures 3.2 and 3.3).

figure 3.2: Automatic evacuation on land Photograph of descender developed by the author under use test

Image removed

figure 3.3 : Automatic evacuation at sea Photograph courtesy by Viking Life Saving UK ltd.(n.d)

When designed correctly, the protection of the person during evacuation is paramount and the more data that can be drawn upon to help facilitate this outcome the safer it is for not only the user, but also for rescue teams and other individuals who may become caught in the evacuation process. Lessons learned from the disasters such as Piper Alpha (Lord Cullen inquiry -1990) can be drawn upon in design for descending into the water, too fast and the risk of secondary issues such as premature deployment of the life preserver, too slow and the evacuee is put at risk of suspension in a potential danger zone such as flash fire or explosion.

The proposed research concept described here is based on automatic response together with simplified mechanical mechanisms that are quite tolerant to external changes and different weights.

Furthermore, the study looks at materials and construction to minimise weight and improve portability, which when included in the design outline, improve the device ability to meet or overcome potential hazards. The action of the design should be to fool-proof and suitable for adults, children and the disabled.

To support the investigation of a comprehensive design for the evacuation device several test rigs were developed and a number of designs were manufactured for analysis and full function testing with a specially commissioned articulating manikin (figure 3.6), Torso (EN-1992 (figure 3.4) and ANSI/CSA-2007 (figure 3.5)), and weights to fully understand visual performance.

figure 3.4 : CE Torso - 100 kg Photographed by Author as new torso acquired for testing

figure 3.5 : ANSI/CSA Torso 100 to 160 kg Photographed by Author as new torso acquired for testing

Image removed

figure 3.6 : Articulating manikin 100 kg, Full jointed skeleton with a rubber skin, photograph provided by Ogle prior to delivery.2009

3.3 Rescue

The second criteria for a descent device is rescue and the ability to move and deploy a unit to assist a casualty or an individual who is in a precarious or dangerous position. The proposed research concepts examined here take into account this requirement and the data and information gained by this research is to assist fellow engineers and designers in finding solutions to potential problems, special rescue dummies were manufactured for trials with the designs as shown in figure 3.7. The introduction of descent controllers in rescue and evacuation would enable the protection of/and preservation of lives. The interaction between the design and the individuals is such that in a moment of severe stress the design is easy to operate by even the lowest skilled individuals. This study will put forward designs and evaluate them providing data. Furthermore, during the course of this study several body devices were developed and tested with the descender designs using test rescue dummies that were constructed specifically for this research

figure 3.7 : Rescue dummies - 30 kg and 100kg Photograph by author of special manufactured dummies to authors requirements during testing

figure 3.8 : Rescue sling - body device - by author

figure 3.9: Rescue triangle - body device for evacuation and rescue by author

3.4 Interdependence between the individual and descent design

There is an interdependence between the individual who is using the device and the device itself. If misused, even the most simple designs would have the potential to fail.The author has developed designs based on the minimum requirement for training in addition to covering the broadest range of applications.
Furthermore, environmental influences such as fire, explosion, height, weight, disability, collapse of structures, ground condition or type, and weather all add to and define the descent design that can be adopted.
The final part of the descent process is the body device two designs are given above - figures 3.8 & 3.9. The use of any design is dependent upon its ease of use and the design of the body device or harness, which has to be able to fit and meet the individuals needs for height, weight, ease of donning and importantly confidence in order for the person to use the device and for the descent design to perform correctly.

Chapter 4 : Lifeline

4.1 Introduction

This chapter provides considerations given to the lifeline type that can be used in the design. The lifeline is a critical factor within any design as it is the part that connects the individual to the descent mechanism. It does, however, have an interaction role with the descent design as it determines to a lesser or greater extent the size of the final design.

Based on construction, material and shape, the lifeline provides the motion for the descender as no external power source is to be used. The lifeline is also the item along with the body device that the individual will be most aware of during any descent. Hence it has the additional factor of being capable of providing a level of confidence for any solution.

The lifeline falls into three main categories, namely webbing, fibre rope or wire rope.Due to the nature of high rise developments one criteria is height greater than 35 m, but up to and beyond anticipated100 m. Wire due to its stiffness, construction, weight and diameter (greater than 4.1 mm) required to meet minimum static rating of 12 kN (EN341 1992), have for lengths exceeding 20 m been put aside for the purpose of this research.

4.2 Webbing

4.2.1 Introduction

Webbing due to its flat profile is considered to have potential in meeting the requirement for a compact descent control device. The lifeline has to be able to survive under adverse conditions. Therefore, material selection is important and in using narrow fabric otherwise referred to as tape, has the ability to maximise capacity in a small housing. However, tape has the disadvantage of not reeling back very well. It tends to fold and it also presents a higher frontal profile to incident wind in comparison to rope or wire. In selecting webbing as a potential lifeline for this research,the aim was to not only concentrate on its thickness but also to determine how narrow it could be whilst trying to achieve a 15 kN terminated static strength, although standards point wt 12 kN as the requirement the higher level would enable multi use designs with fall arrest in accordance with EN 360 2001 in case a back up fall arrest brake is employed.

In 1989 R.H Ericksen together with R Koch (SAND-88-2592C;CONF-8904118-5;1989 updated 2008) developed a kevlar parachute tape woven with a flat then tubular then flat construction. The purpose being to reduce the weight of the suspension lines whilst providing higher strength, they achieved an 8 % reduction in Weight with an increase in strength of more than 10%. When considering the construction and type of weave for the descender the aims of Ericksen and Koch vary to the aims of this research in so much as the tape has to be compacted for maximum length into a confined space, on deployment twist, reeling capability and compactness are features considered more relevant. In the paper it is considered that the new tape can be manufactured to very tight tolerances which again if used for a number of parachute suspension lines is a benefit, however, in the case of a single line descender this advantage is less relevant.

In reviewing the availability of tapes and webbing, for the lifeline, Daniel Gallucci (High strength multi layered tape; US patent 5,254,387;1993) considers and refers to the use of multiple ribbons, joined together with flexible thermal resin, for attachment to structures and other products (figure 4.1).
The aim being to replace or co exist with other conventional ways of reinforcing buildings, and structures such as rebar. The tape offers perhaps a more cost effective way of increasing structural strength and integrity. As with the previous reference the issue with using multi layered tapes, would be one of compactness and thickness. The ability to be wound into small confined space whilst still retaining a long descent length. Unlike the previous reference it is considered that this would add weight with very little benefit.

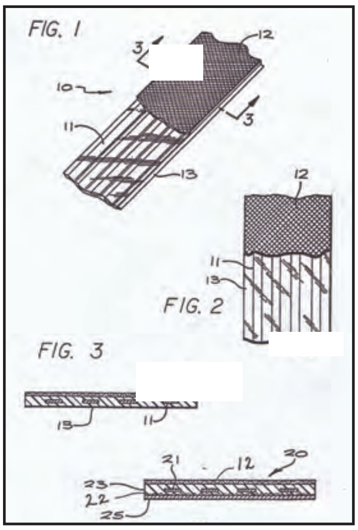


figure 4.1 : Multi layer webbing with resin binders. US Patent 5,254,387:1993

4.2.2 Construction

The webbing strength, thickness and width were considered and several options were manufactured for trial. The material of construction chosen early on in the study was Kevlar. Kevlar type 29 has a number of positive characteristics in that it can be woven to produce a very high strength tape of minimal thickness also it has the added characteristic of not burning. The tape starts to decompose at elevated temperatures above 600 ^{0} C, which is outside the operating parameter for the designs proposed. Also considered was the weave pattern. After consultation with the weavers it was considered that a weave which was made up of bands similar to that on the seat belt in a motor vehicle would add torsional stiffness to the tape, resisting spin when in use in addition to a higher resistance to wear and tear. One of the characteristics that the Kevlar had that required consideration was the effect of ultra violet rays on the material. If used it would have to be shielded from daylight until such time as it was deployed.

Two main webbing weave constructions were manufactured as shown in figures 4.3 and 4.4. The band weave or herringbone weave has a series of panels that offer torsional stiffness to webbing and improved wear resistance. The samples were prepared for tests and tensile tested, figure 4.2 shows samples after testing.

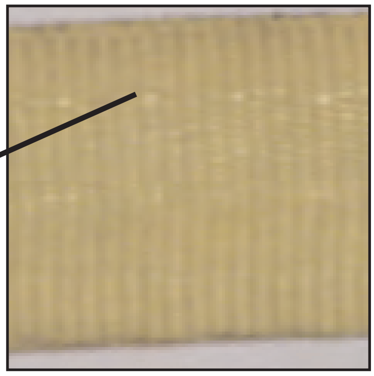


figure 4.3 : Kevlar test samples Band weave

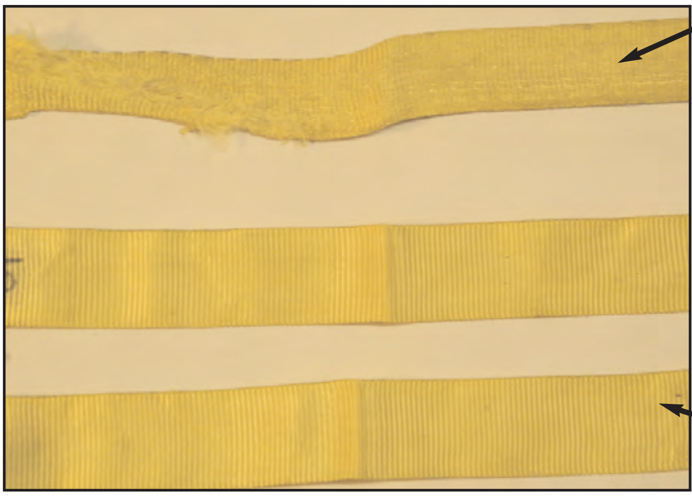


figure 4.2 : Kevlar test samples

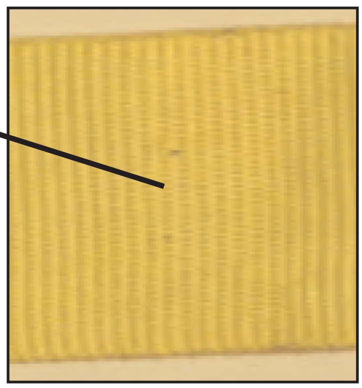


figure 4.4 : Kevlar test samples Straight weave

4.2.3 Termination

The tape, when produced in Kevlar, has a slippy appearance which makes termination problematic. The material was found to creep and move whilst sewing, which meant obtaining samples for test could only be achieved using special clamps that had to designed for use on a modified sewing machine. A Brother 311 E (figure 4.5) sewing machine was modified to accept special clamps as shown in figure 4.6 in order to clamp the webbing whilst being sewn.

figure 4.5 : Modified sewing machine centre showing special clamps used to produce the test samples

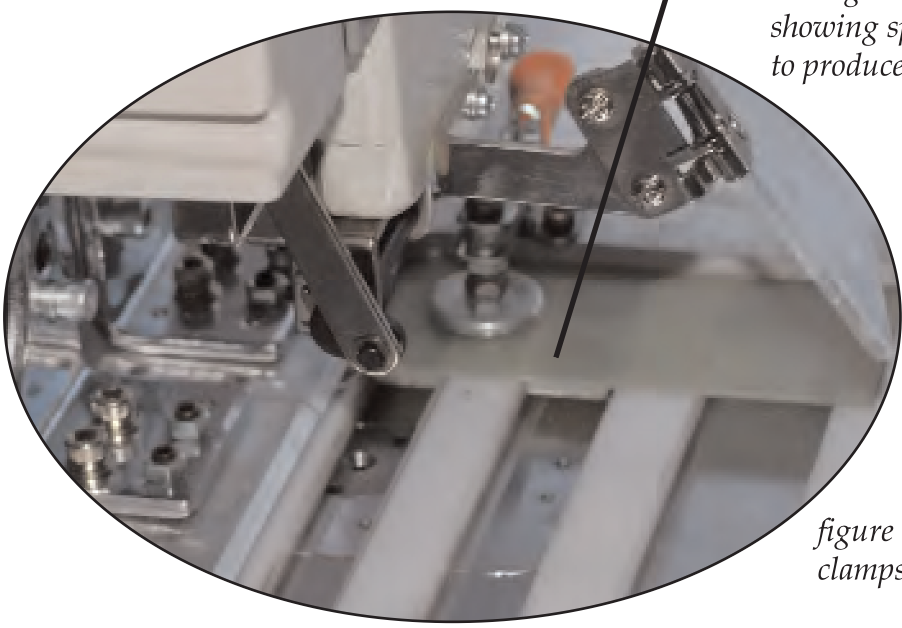


figure 4.6 : Special clamps enlarged

In order to produce the samples for testing with the designs and to prove their static strength, special needles had to be experimented with. Kevlar is a very difficult material to cut or sew. Therefore Teflon coated needles were required with round tips in order that the machine could sew the Kevlar together and produce looped terminations. A number of stitch patterns (examples are shown in figures 4.7 to 4.9 inclusive) were investigated to see which would allow for the slip between the two pieces of Kevlar and be able to achieve the strength required 15 kN.

figure 4.7 : Box and gate termination after testing

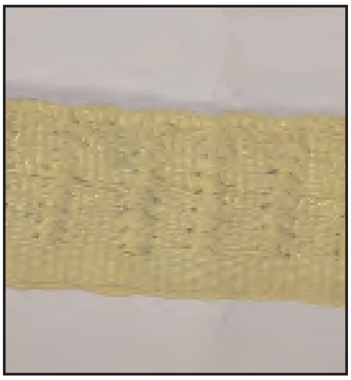


figure 4.8 : Bar tacks in 6, 10 and 15 tacks after testing

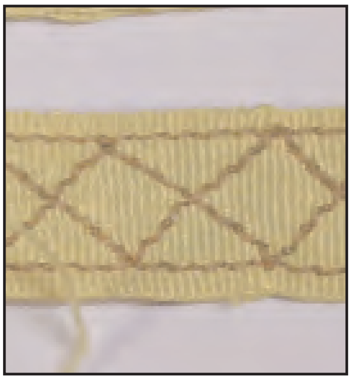


figure 4.9 : Box with multiple gates

The securing thread was also considered with trials using a number of materials in order to determine the best material for the design. Polyester, Nylon, Kevlar and Zylon threads were used. The Zylon thread was specifically manufactured for this study as it was not commercially available. The thread had to have a number of characteristics such as strength and the ability to survive potential heat or fire hazards.

4.3 Rope
4.3.1 Introduction

Many ropes were considered for the study, however a special rope used in mountaineering and pot holing was considered to have the best construction to meet the 12 kN (EN 341 1992) and size considerations of any design. With low stretch the mountaineering static rope which could be manufactured with materials such as Kevlar or alternatively manufactured with a thin steel core in stainless steel or galvanised steel. The rope with steel core had been developed as an escape cord for use with the original descenders. However, with the recent developments of flame retardant materials and the stiffness of the cord it now offered very little advantage to the new kernmantle ropes.

The rope that could be incorporated requires to be of flexible construction and able to withstand bending over long periods and action. R.E Hobbs and C.J. Burgoyne carried out research into bending fatigue in high strength ropes. The paper related to parallel lay or Parafil ropes using kevlar 49 and kevlar 29 fibres. Figure 4.10 shows the layout of one of the tension rigs for straight pulling, others used were for 45 and 180 degree bends.

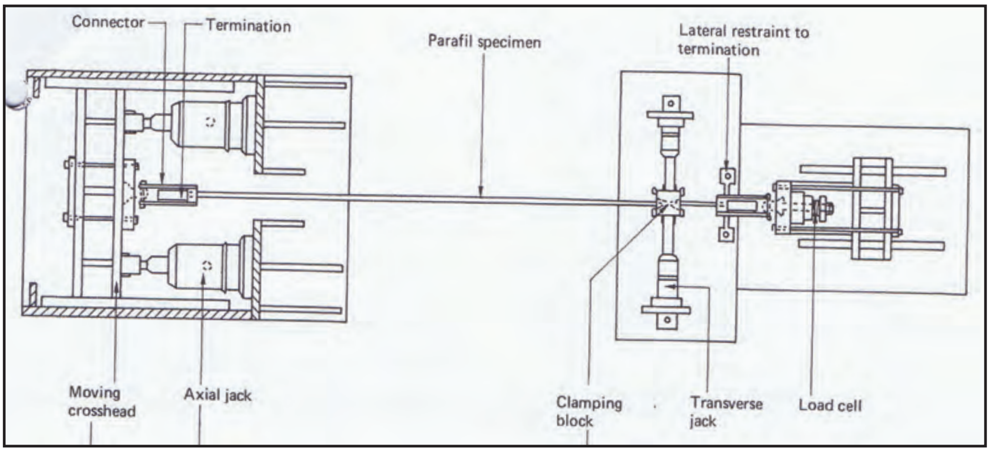


figure 4.10 : Tensile test machine. (R.E.Hobbs;C.J. Burgoyne Bending fatigue on high strength fibre ropes, 1991)

The conclusions found were that as the bend angle increased the number of cycles until failure had as expected reduced to around one third of the cycles at 180 Degree as compared to straight tensile. The unusual factor that they found related to where the failure occurred, it had been assumed that failure would be at the socketing or termination point, but in fact this proved not to be the case with failures following a more unpredictable path.

In tests carried out under this research with kern mantle construction failure was found to occur at the termination, as the main fibres were not kevlar as in the previous case then the findings have less bearing unless Aramid fibres were to be used in the descent rope.

The rope chosen is a braided sheath using a twisted core mantle, Henry J. Holzhauer patented his findings on high strength, low stretch braided ropes (US Patent 3,968,725:1976). It is noted that the combination of high module core fibres and lower module sheath fibres provide strength via the core and abrasion resistance due to the sheath. by combing fibres in this manner the result is a more stable rope able to cope with different environments. Static mountaineering ropes take advantage of this fact in order to produce better wearing ropes capable of multiple use. The ropes are constructed to reduce sheath slippage in order to permit amongst other matters a better termination and improved performance. This has advantages for descent equipment which may rely upon contact or driving of the sheath but wishing not to bunch up as the rope passes through the unit.

In figure 4.11 A graph is presented showing the elongation of a braided rope as an applied load is introduced. The composite rope giving the lower elongation. With descenders it is considered better to have low stretch for both the performance of the unit as stretch would reduce the rope diameter and in order to have predicted drop distances.

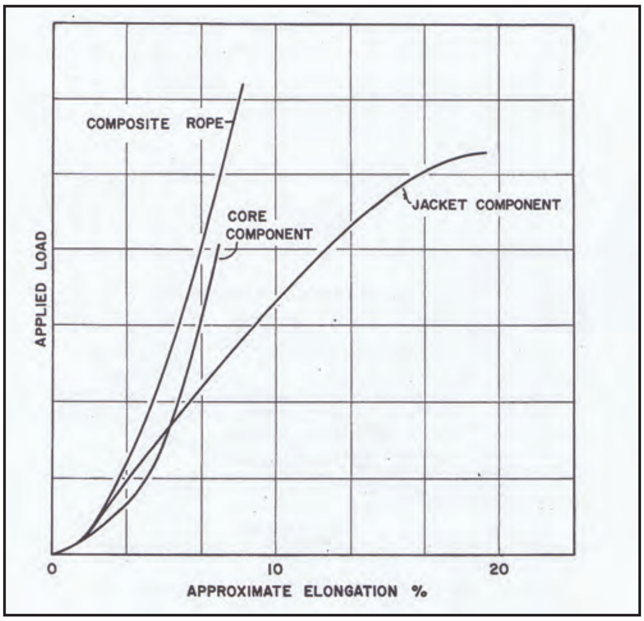


figure 4.11 : Elongation v's Applied load . High strength,low stretch braided rope; Henry J Holzhauer; US 3,968,725:1976

J. Gladenbeck and G. Muller carried out research into high strength rope filing US patent 4,022,010 in 1977. In this paper they consider the rope to comprise a jacket with several core strands or components that them selves could be in jackets. A plastic material or rubber type compound could be used to jacket the parts in a parallel core arrangement.

The resultant would give high strength but is considered to be too rigid for descenders and in the case of drum driven units the sheath or jacket is required for the unit to grip and work (reliant upon the sheath weave picks to grip). Figure 4.12 shows the construction that was proposed in the patent. This type of rope would work in horizontal life lines or cable pulling/laying on land or at sea where strength durability and resistance to the environment are more relevant than required for personal evacuation devices

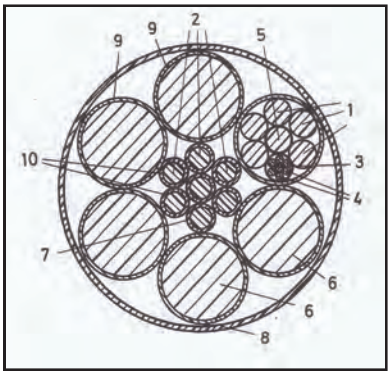


figure 4.12 : Rope cross section construction. High strength rope; J Gladenbeck and G Muller: US4,022,010:1977

4.3.2 Construction

After consideration of wear and size for the purpose of achieving the aims and objectives the construction chosen was kernmantle, where the mantle is the sheath of the rope of braided construction (figure 4.13) which protects the core or kern, the kern being made up of several strand that are twisted together.

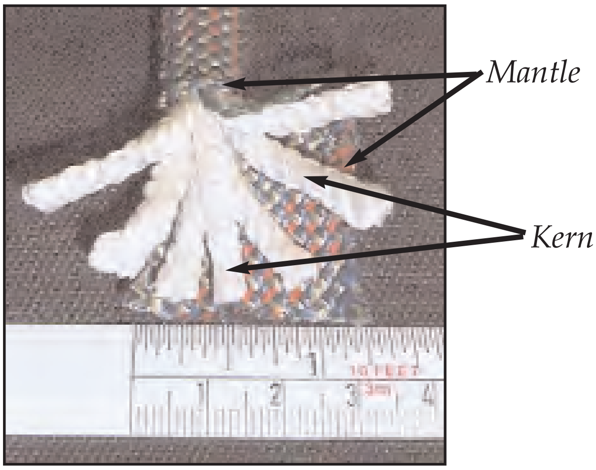


figure 4.13 : Kernmantle rope - construction

Typical breaking strength for a 9 mm static rope is 21 kN (tensile tests carried out clamped) with a weight of 51 g/m being typical. This provides a lightweight rope that can coil easily and yet still achieve the strength requirement.

4.3.3 Termination

The fibre rope was terminated by swaging and sewing, both methods maintain a high strength for the terminated rope without damaging it. Swaging involved three or four swages being applied to the rope with a distance of approximately 50 mm between swages. This was to allow for any slippage and also to prevent damage to the rope.

Brother Corporation developed a sewn termination for use with their machining centres. Special clamps had to be developed and manufactured in order to sew the termination which used size 40 thread at very low sewing speed, the higher the thread number the finer the thread diameter, for example to sew webbing size 18 or 20 would be used.In order to sew the rope and get between the rope icks then the thinner the thread size.

In figures 4.14 and 4.16 the swage termination and press is shown, where as in figures 4.15 and 4.17 the sewn termination is shown.

figure 4.14 : Kernmantle in swaging press

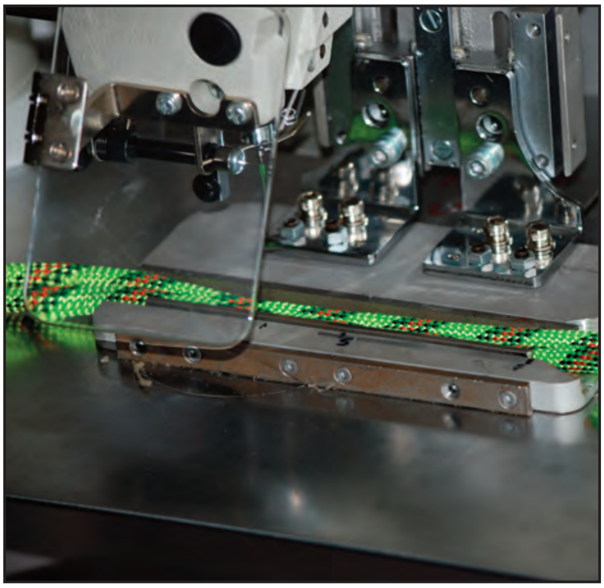


figure 4.15 : Kernmantle in special clamp on sewing machine

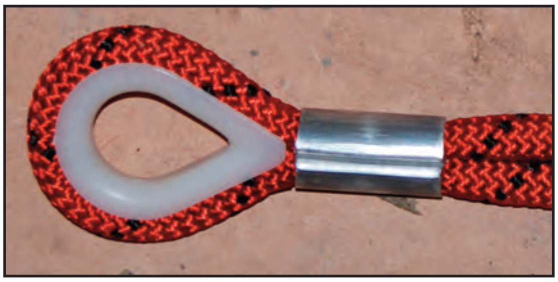


figure 4.16 : Kernmantle swaged termination

figure 4.17 : Kernmantle sewn termination

4.4 Tape

4.4.1 Introduction

In order to facilitate the maximum descent, whilst maintaining a lightweight solution a web weave and construction had to be determined. With no known commercial tape available which fits the thickness, width and strength requirements available, the study and research of this aspect required exhaustive investigation with the manufacturer (Ryknel Tean,Derby)of tapes and the development of test protocols to determine if a tape could be used in the design. The tape size and termination were both unknown quantities.

As opposed to the rope no information or prior art existed. This section deals with how the webbing samples were tested and then deals with the termination.

4.4.2 Tape type and tensile testing

A number of tapes were manufactured for the research programme, in order to determine the tape strength. Prior to termination samples were placed into a tensile machine with the ends clamped in order to prevent slippage, the load was increased until failure.

Two webbing types were manufactured and investigated (i) a herringbone weave with 4 panels (figure .) and (ii) a straight weave single panel (figure 4.19).

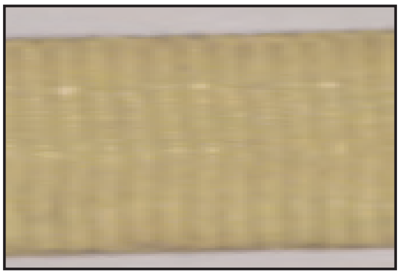


figure 4.18 : Herring-bone - 4 panel weave

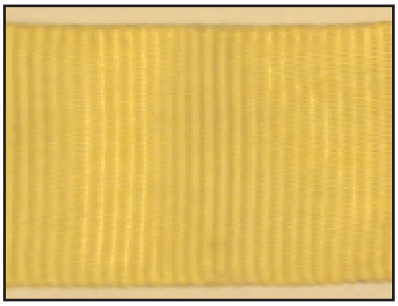


figure 4.19 : Straight weave - single panel

The two weave patterns were produced in two widths and two gauges. In the herringbone weave the two sizes were 20 mm wide by 0.6 mm thick and 25 mm wide by 0.7 mm thick whereas in the straight weave the samples were manufactured 25 mm wide by 0.65 mm thick and 22 mm wide by 0.45 mm thick.

To determine the compensation required to apply to the results a rigid link was inserted and tested. A Hounsfield testing machine was used and in order to allow for the load cell deflection at one end of the machine a rigid link was inserted and tested *(*figure 4.20*)*. The results are tabulated in Table 2 together with values for stress,strain and E modulus

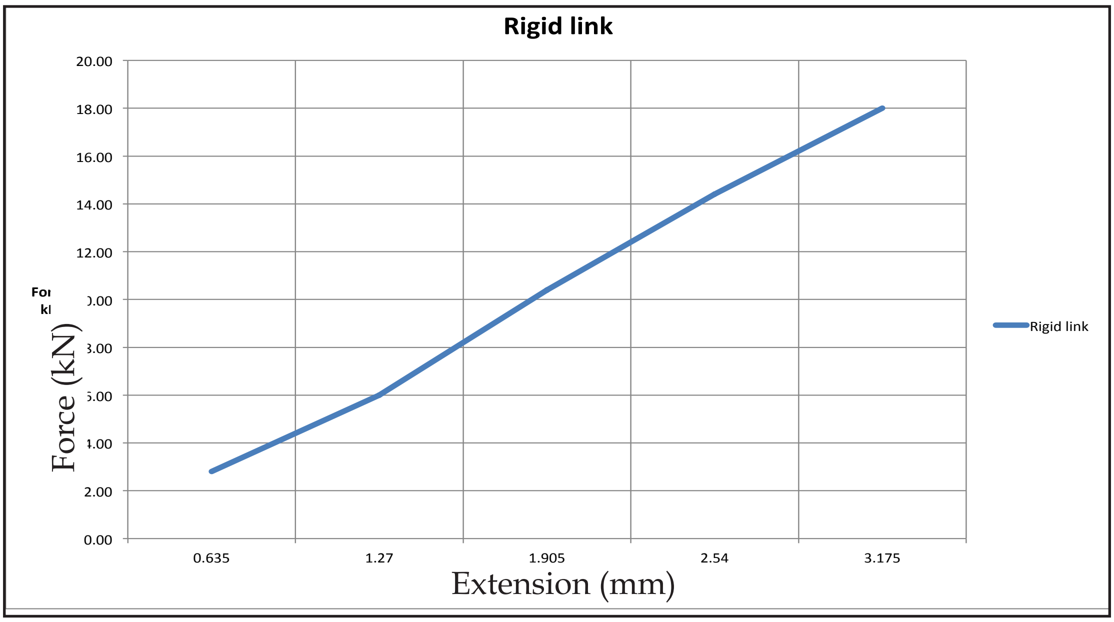


figure 4.20 : Rigid link compensation test

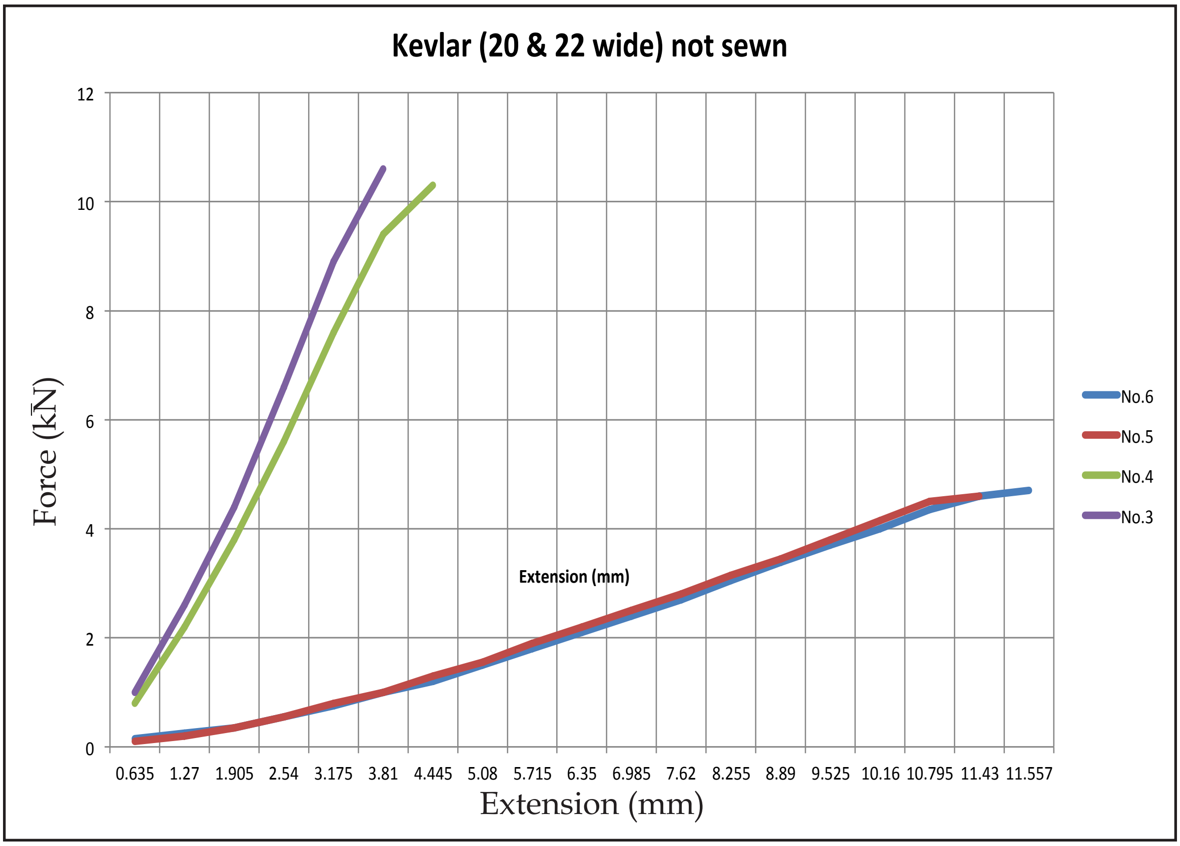


figure 4.21 : Tape test without termination

The straight weave samples failed at 4.6 kN (figure 4.21), whereas the herringbone samples achieved slightly over 10 kN. However the target strength of 15 kN was not achieved with any of the samples. Values were obtained for Youngs modulus E, stress and strain (tape size provided by Rykneld tean of Derby and checked upon receipt).

Figure 4.22 shows a photograph of the tapes after tensile testing.

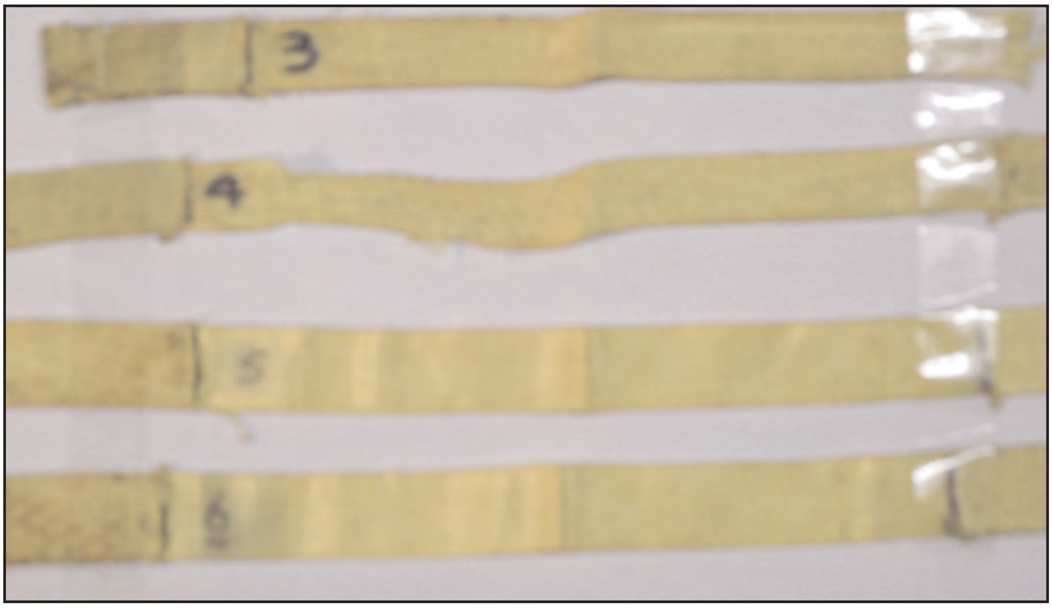


figure 4.22 : Photograph of test samples after the test

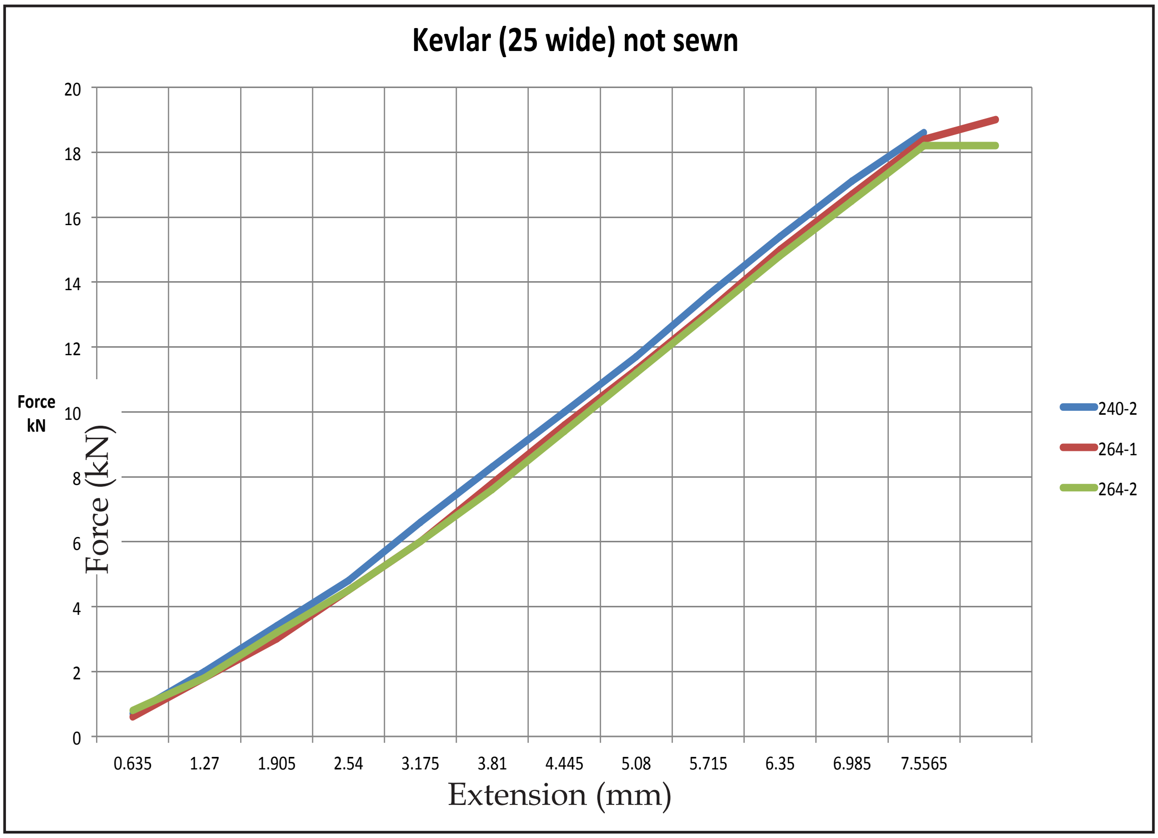


figure 4.23 : Increased tape width tests

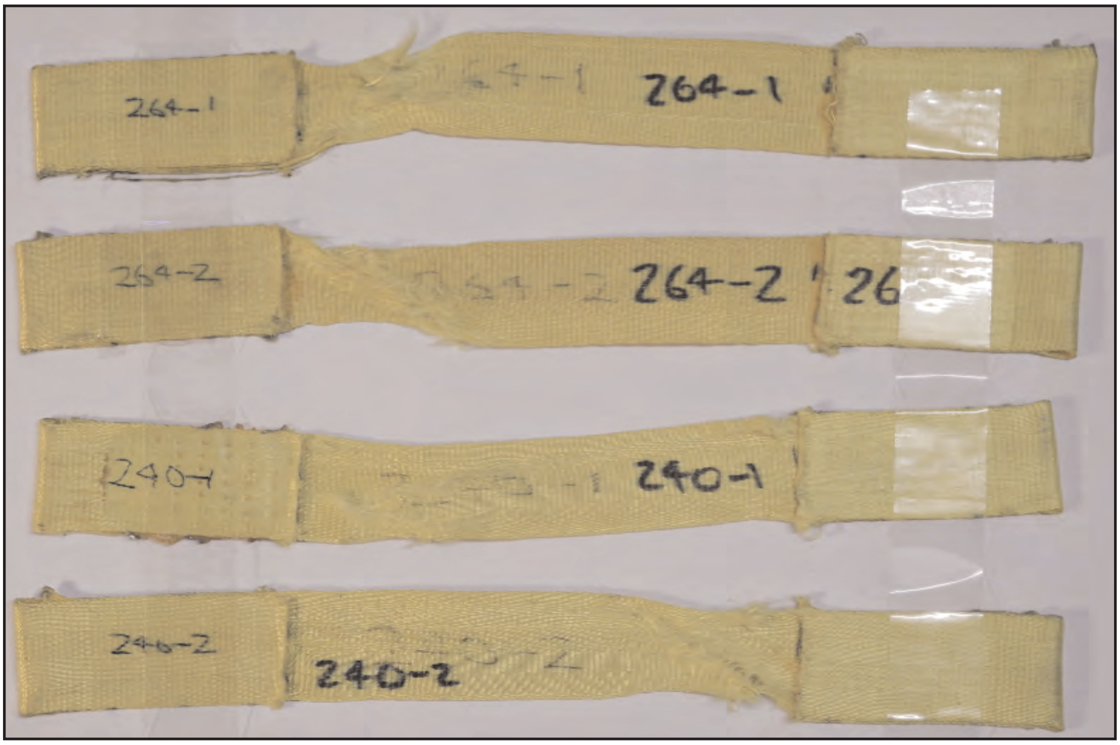


figure 4.24 : Increased tape width test photograph of samples after test

The table tabulates values from figure 4.20 and figure 4.23 to determine stress strain and E modulus for the tape. From figure 4.20 and figure 4.23 by increasing the width to 25 mm samples 240 and 264 the strength of the webbing exceeded the 15 kN level with break occurring at over 18 kN. The width and thickness of the samples was verified by the supplier Rykneld Tean from their laboratory results enabling a value for E to be determined. Figure 4.24 is a photograph of tapes after tensile testing.

Description	Tape width	Tape thk's	Test length	Force	Extension	Extension corrected	Strain	Stress	E Modulus
	mm	mm	mm	kN	mm	mm		N/mm2	N/mm2
Rigid link	n/a	n/a	n/a	10	1.59				
kevlar 264/2	25	0.7	100	10	3.68	2.09	2.09E-02	571	2.73E+04
kevlar 264/1	25	0.7	100	10	3.68	2.09	2.09E-02	571	2.73E+04
kevlar 240/2	25	0.65	100	10	3.62	2.03	2.03E-02	615.4	3.03E+04
kevlar 240/1	25	0.65	100	10	3.75	2.16	2.16E-02	615.4	2.85E+04
kevlar 4H -3	20	0.6	200	4.5	2.5		1.25E-02	375	3.00E+04
kevlar 4H -4	20	0.6	200	3.4	2.5		1.25E-02	283	2.26E+04
kevlar - 5	22	0.45	200	1.3	2.5		1.25E-02	131	1.05E+04
keblar - 6	22	0.45	200	1.2	2.5		1.25E-02	121	9.70E+03

Table 4.1 : Chart of webbing stress/strain and E values obtained from the test

4.4.3 Tape termination introduction

The next phase of the lifeline development was to terminate the tape such that the 15 kN level could still be achieved. The use of polyester thread failed as the Kevlar would simply shear through the thread under low loadings. Three thread types were investigated Nylon, Kevlar and Zylon.

The results are presented in figures 4.25 ,4.26, 4.27, 4.28, 4.29 and 4.30. The Zylon thread was an experiment and trials were performed with Oxley Threads in attendance as little was known about the sewing characteristics or the structure of the thread. As with Kevlar, Zylon decomposes when subjected to temperatures over 600 ^{0} C, but its specification has advantages over Kevlar as it has both better shear and bend radii whilst maintaining an equivalence to Kevlar in longitudinal strength.

4.4.3.1 Termination with Nylon thread

Nylon thread was used in different configurations and the figure 4.25 shows a typical result, the sewing pattern was a box and gate configuration and figure 4.26 shows three of the samples photographed after the test.

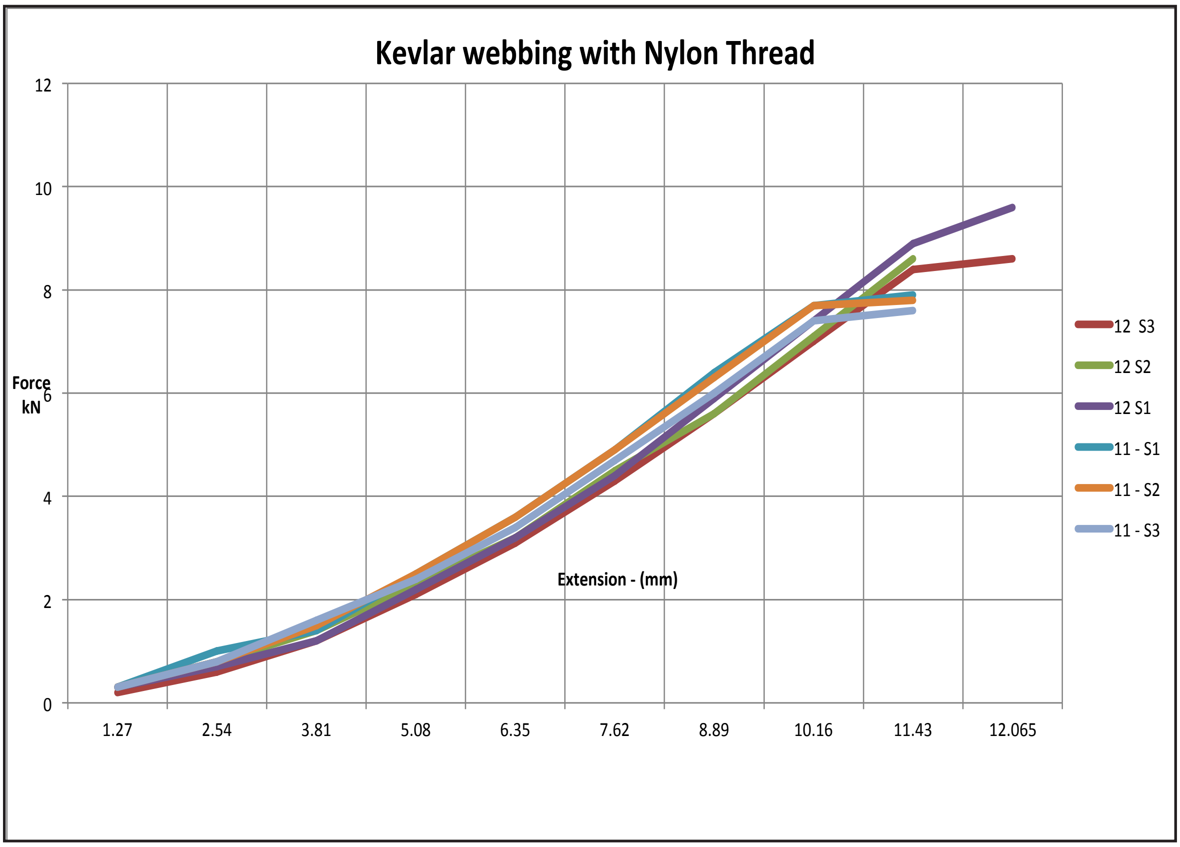


figure 4.25 : Nylon termination test results

figure 4.26 : Photograph of samples after testing

4.4.3.2 Termination with Kevlar thread

Kevlar thread was used in different configurations, figure 4.27 shows a typical result, the sewing pattern was a bar tack configuration with either 10 or 15 bar tacks per termination and figure 4.28 shows four of the samples photographed after the test.

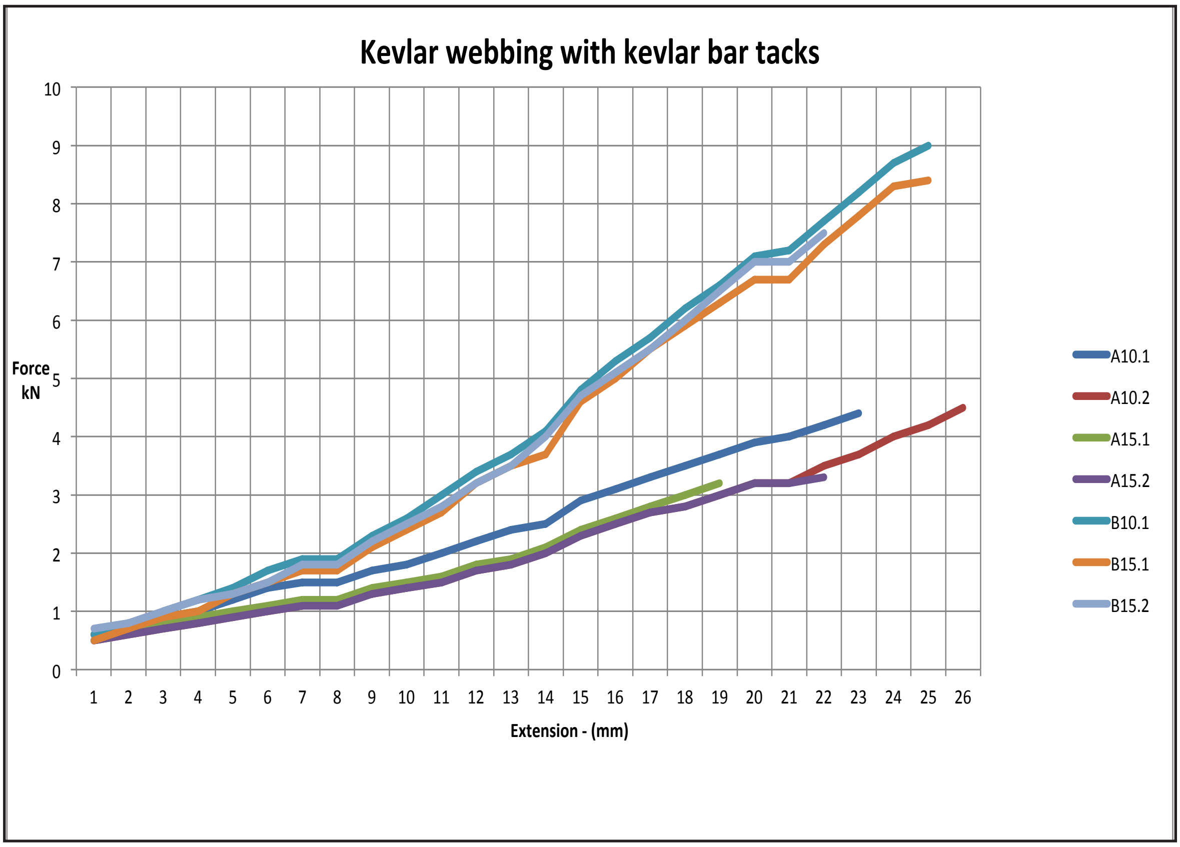


figure 4.27 :Kevlar bar tacks termination test results

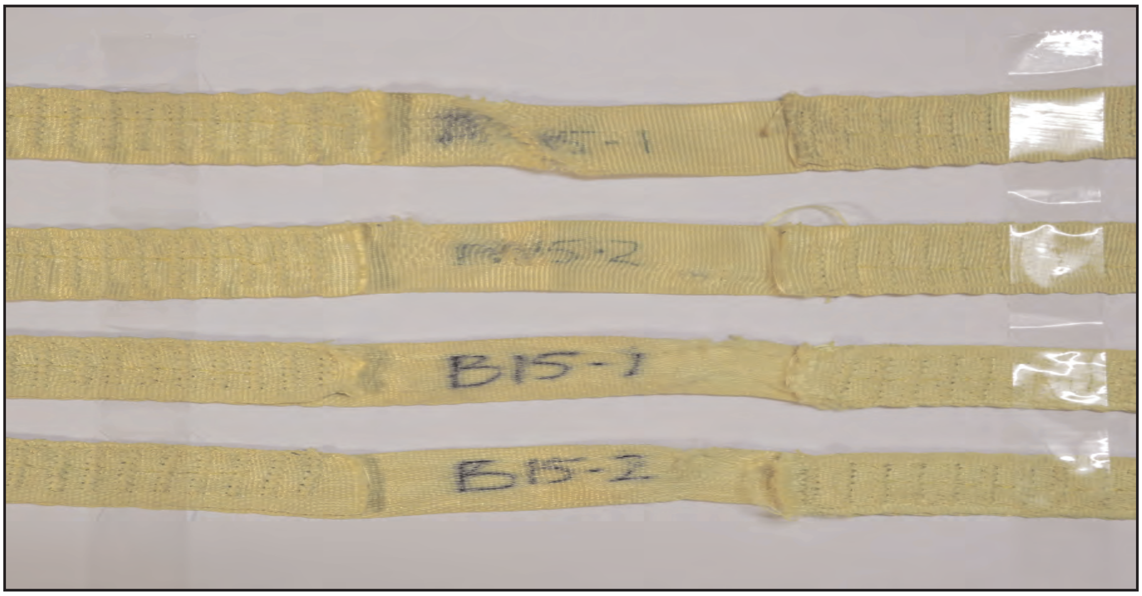


figure 4.28 : Photograph of samples after testing

4.4.3.3 Termination with Zylon thread

Zylon thread was used in different configurations. Figure 4.29 shows typical results. The sewing pattern was a box and gate configuration with 3, 4 and 6 gates. The figure 4.30 shows five of the samples photographed after the test.

figure 4.29 : kevlar with zylon thread termination tests results

figure 4.30 : Photograph of samples after testing

4.4.3.4 Termination and webbing discussion

The narrow webbing at 20 mm and 22 mm wide failed to meet the strength requirements that was required. Also there is a significant strength reduction between the herringbone 4 panel weave pattern and the straight weave during the first narrow tape tests with the herringbone weave producing results almost twice that of the straight weave at 10 and 4.5 kN respectively. However, the two weaves produced repeatable results.

When the 25 mm wide tapes were tested the results show a significant increase in strength together with matched results irrespective of the slight thickness changes from 0.65 to 0.7 mm. The increased width used the herringbone pattern based on the previous weave tests as the herringbone produced the significantly better results.

Polyester thread would not hold any force so was omitted from any further tests with Kevlar simply shearing through the polyester thread.
That left three further options namely:
(i) Nylon (ii) Kevlar and (iii) a new prototype thread not on the market Zylon (appendix3).
The Nylon thread gave repeatable results but the stitching would fail below 10 kN, also with a design that uses Kevlar, then due to environmental concerns flames and heat the use of Nylon as the anchorage termination would be a strange choice even with shielding based on its performance it would meet the single use criteria of 5 kN but not the desired 15 kN.

Prior to the tests it had been assumed that the Kevlar thread would provide the desired result and would not have the environmental concerns of Nylon. However, under test the Kevlar thread is unpredictable and the termination strength results were poor. Kevlar demonstrating a fundamental weakness, which is its inability to withstand shear or bending, whereas, in comparison the Nylon would allow movement and stretch so producing repeatable results, the Kevlar would shear the stitches and fail. The sewing of Kevlar is quite difficult and automatic machines do not like trimming (automatic action by the machine to cut the thread at the sewing clamps, releasing the sewn part) as it blunts the machines cutters very quickly.

Zylon on the other hand shares many advantages with Kevlar but its main advantages are in bending and shear. The box patterns suited the combination and readings in excess of 15 kN were achieved. In fact the results do not show the full picture as the webbing or stitches did not fail but exceeded the test rig measuring limit of 18 kN.

The lowest value was achieved using a 22 mm wide straight tape that achieved the same results terminated as unterminated which indicates that the termination method had no detrimental effect on the tape.The slightly lower results for two tapes was due to them being 20 mm wide herringbone weave which achieved higher values terminated than no terminated. Again the 25 mm webbing with herringbone pattern proved the best tape and the Zylon thread combination worked well.

Chapter 5 : Evaluation of lifelines

5.1 Wire rope lifeline (figure 5.1)

The descent control design can continue along several routes. If the height is less than 20 m height it may be advantageous to utilise a wire rope as the lifeline. This could develop into a self rewinding design for multiple evacuation or recreation. With no trailing lifeline and the ability to fast rewind due to the selection of wire rather than tape or fibre rope this type of design would cover a number of potential risks or evacuation needs.

In order to achieve the minimum tensile requirements for designs for mass evacuations the wire would have to achieve 12 kN in tensile strength in order to meet the current and envisaged International standards that are written or being developed subject of PrEN.

Wire, depending upon construction would require a minimum bend radius of between 10 and 20 (ratio i.e. 10 d to 20 d where d is the diameter of the wire rope in mm) as a multiple of the diameter. The wire diameter should be kept as small as possible and from a review of wire rope supply data(Bruntons of Scotland guide used) this indicates a diameter of 4.1 mm (nominal 5 mm) would be required. In order to keep the design as compact as possible the construction of the wire would also be important, reducing the size of the drum required and also reducing the internal friction which would impact upon the rewind mechanism. From studying the supply data from several wire manufacturers the optimum construction for the lifeline would be 6 x 19 (six core with 19 strands of wire per core) with a fibre core as this would allow a minimum bend radius of 10 times the diameter to be applied.

If the design were to move to a single use with no rewind, then the wire rope diameter could be decreased to 3 mm as the strength requirement for such designs set by the International standards is 5 kN. With the reduced diameter and continuing the minimum bend radius of 10 times the wire rope diameter the design would either be smaller and more compact or the design would be capable of holding more lifeline permitting greater descent heights to be achieved. Also the lack of a rewind mechanism would assist the design size and shape.

The other disadvantage with designs that use wire rope is its weight.

Compared to fibre rope a 5 mm diameter wire rope weighs typically 90 g/m compared with 9 mm diameter fibre rope 50 g/m and tape which typically weighs 8 g/m

Each design of descender represents the culmination of engineering detailing in order to achieve the result., in appendix 5 the main points of figure 5.1 are given in detail design drawing format.

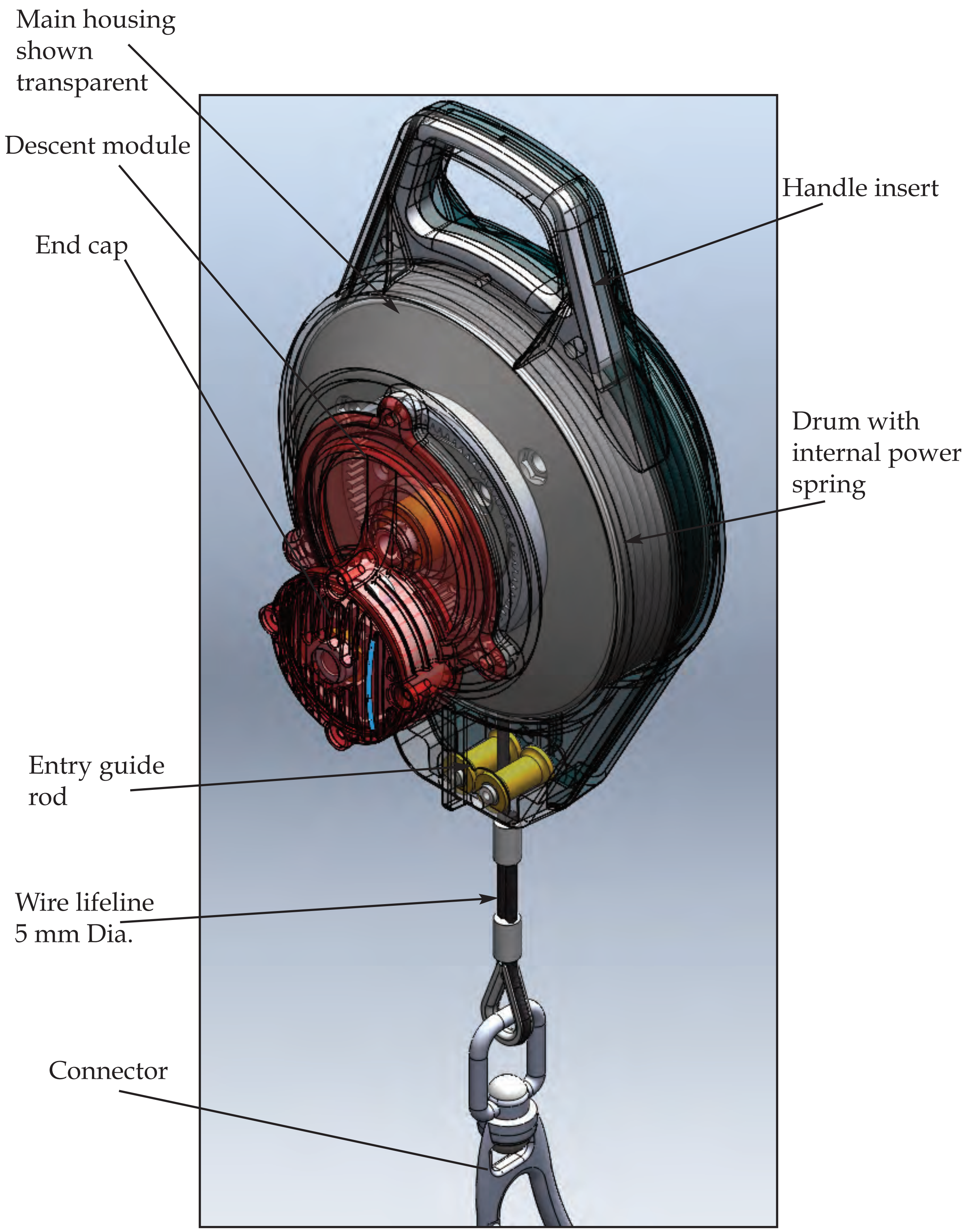


figure 5.1 : wire rope descender developed by the author as part of this research programme

5.2 Fibre rope lifeline (figure 5.2)

This section discusses the descent control design developed with a fibre rope as the lifeline.

This type of lifeline of kernmantle construction has been used in rock climbing and abseiling for many years. The rope can be fire retardant and coated to prevent water absorption. A typical construction has a multiple twisted core or kern with a 16 plait polyester cover or mantle. It has a low stretch and is used for fast rappelling or rapid descent such as by the military from helicopters.

With environmental issues that may affect the individual during descent, rope manufacturers have increasingly looked at protecting the kern of the rope with an Aramid material. Materials such as Technora and Twaron have been used which create an abseil rope that can withstand temperatures of up to 500 ^{0} C. These ropes are slightly heavier than standard static ropes with a like for like comparison going from a typical 50 g/ m to 67 g/m which has to be bourne in mind when developing a descent control device if portability is the objective. Fibre lifeline is lighter than an equivalent strength wire rope and retains the high flexibility and stowage properties.

The final fibre rope considered was a crossover between wire and fibre, the fibre to give the grip for the rope in early descent controllers, the wire to provide the strength and also a fire resistant core. As with many crossover developments this one has many drawbacks.

It comes in 8 mm diameter with either stainless steel or galvanised steel core. The construction has a cotton inner to reduce core slip and a polyester cover. Weight increases significantly and now it weighs more than an equivalent wire rope weighing typically 102 g/m. It is not as flexible as wire rope being a single strand core which presents stowage problems. Being an older design for a new application the belief was that in the event of fire or heat the cover may melt but the core would remain In reality as the devices it was designed for relied on the interaction between the cover and a drum to operate the braking in tests carried out to determine a lifeline it was found not to operate safely due to the outer sheath being easily damaged and concerns over its inclusion if the sheath were not there through wear or some other aspect , for example flames.. Compared to the Aramid solution it has little or no use and was as a result not included in any design put forward by this research.

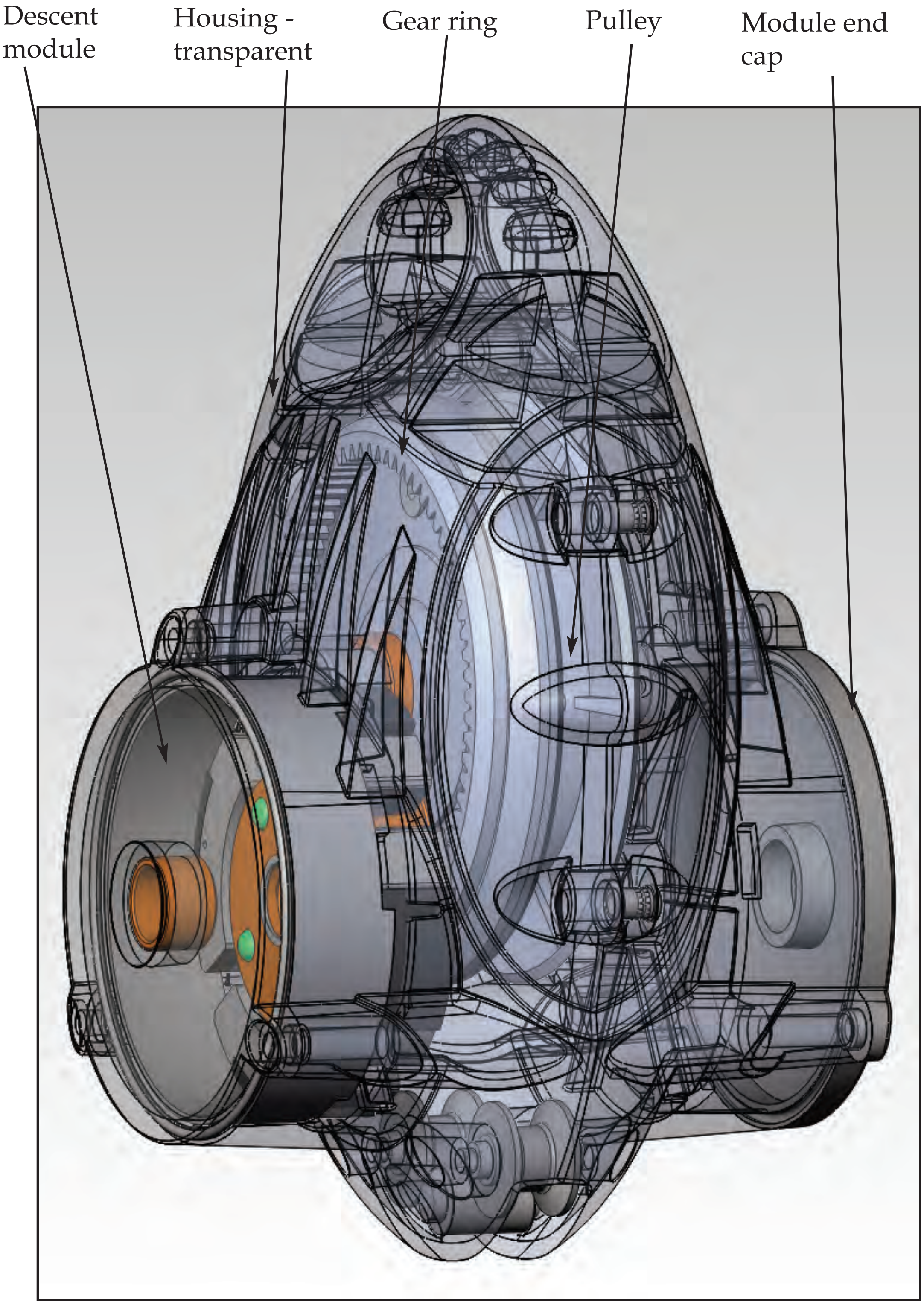


figure 5.2 : fibre rope descender developed by the author as part of this research programme

5.3 Tape lifeline (figure 5.3)

In light of the potential heights that has to be evacuated the use of a tape as the lifeline has many advantages such as very light weight and minimal bend radius for storage and compactness. In chapter 4 the issue over strength and termination were examined with solutions found. The ability to pack 100 m of tape onto a very small drum offers designers another option, moving away from wire rope or fibre rope to provide a lightweight tape unit that is suitable for safe and simple inclusion in a means of descending.

The tape does not rewind easily and the environmental issue of sun degradation have to be considered in any design put forward. However, the tape developed for this study with a thickness of 0.65 mm and a weight of only 8 g/m are significant advances. The termination of tape proved to be a particular problem as convential sewing machine set up and existing threads failed to meet the requirements. The clamps developed for the machining and the Zylon thread (appendix 3)solved the issues encountered and repeatability was achieved.

Further work relating to mass evacuation(chapter 10) is still to be carried out but the tape is considered to be an alternative to wire and fibre ropes due to its inherent advantages.

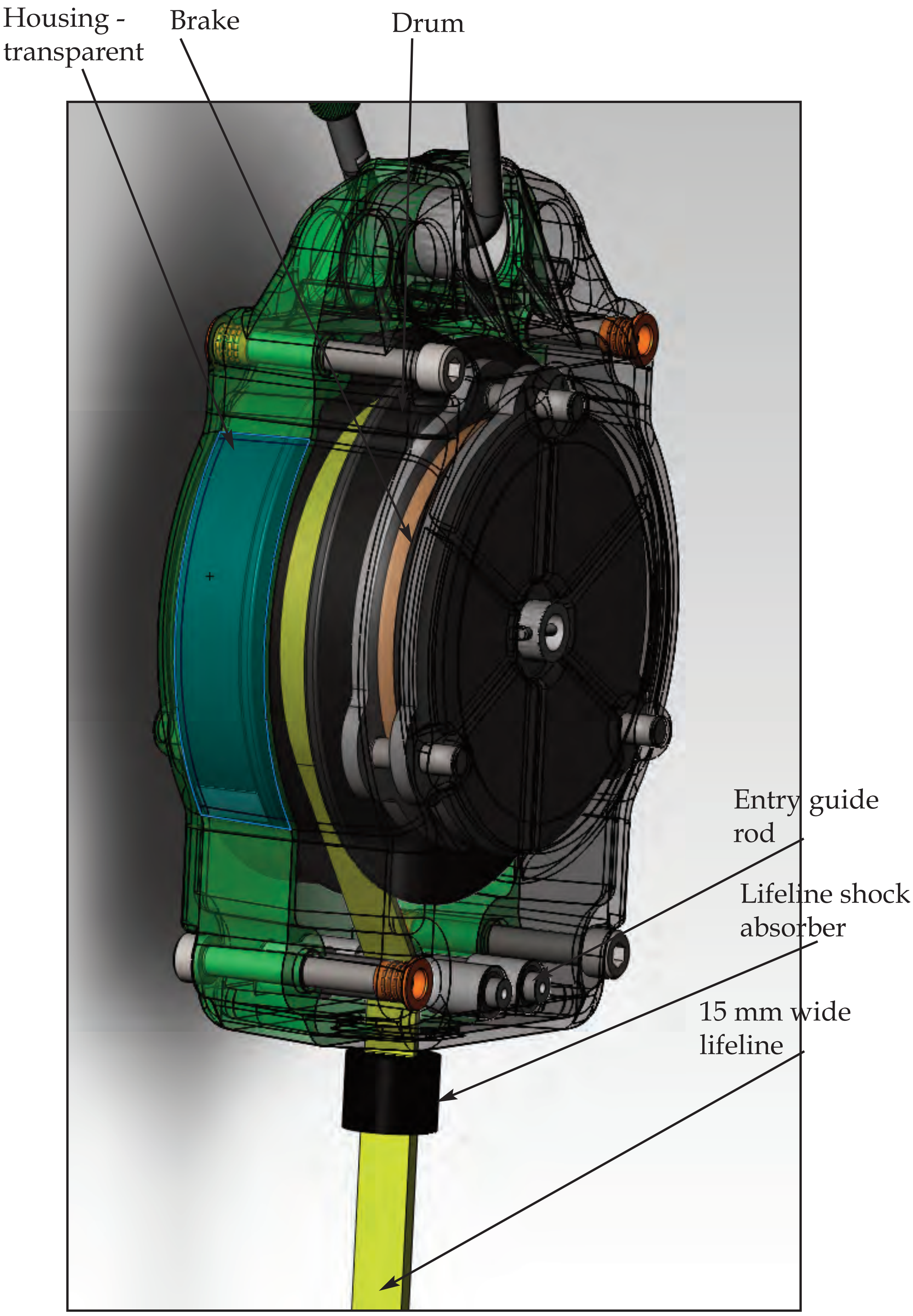


figure 5.3 : tape descender developed by the author as part of this research programme

Chapter 6 : Personal evacuation device concepts and prototypes

6.1 Introduction

In this chapter the principals of design are introduced and then taken from conceptual design to fully functioning prototype manufacture. A number of options were considered and developed, namely:
sliding shoe brake or pivoting brakes employing 1to 4 shoes with leading and trailing shoe configurations.
The design looked at simple mechanics as a method of braking to try and meet the reliability aspect and to keep the number of parts to a minimum. International standards require a maximum on land descent speed of 2 m/s, with an offshore descent speed of 4 m/s, although this is not specified or referenced directly in any standard.
This chapter also considers three type of descender design, the wire rope for up to 20 m, the fibre rope for up to 100 m and the tape from 20 to 150 m. With regard to the upper height limit of both rope and tape, these are considered to be practical limits, but are not considered to be the maximum that is attainable for any particular design.
The design weight of any proposed solution has to be taken into account. Although not legislated by International standards, this is an issue for the designer to address. Considering that the weight of rope for 100 m would be typically 6 kg, if one adds that to the rest of the design the weight would soon become unmanageable for rescue or portability. The devices are, therefore, designed and engineered for minimal weight, whilst still achieving the 12 kN static strength as required by the International standards. The objective of this design is to provide a single brake solution that could be applied to all of the three structure options and be able to achieve the 2 m/s for a weight range of 30 to 150 kg. In each case the brake has to be driven by a drum or pulley.
The solutions stated in this chapter and developed in subsequent chapters have to be able to operate in varied environmental conditions and also have to be able to operate numerous times with little degradation of performance. If the device has to operate in a reciprocating manner, then clearly the brake has to perform in both directions which does have implications as to the configuration of the brake shoes.

Clearly by reversing the direction of the drum or pulley rotation, if the brake shoes are pivoting, then they can move from leading to trailing with subsequent loss of brake torque. In certain circumstances however the configuration and use may benefit from this switch over as for example it may assist the design to rewind the lifeline back into the housing with less torque being developed during rewinding, alternatively it would operate the brake in both directions of drum rotation which could be used in a reciprocating design. All these points are researched in order to identify possible advantages and disadvantages of the designs.

6.2 Construction

6.2.1 Arrangement of parts

The drive for the braking mechanism consists of a drum or pulley, to which a gear ring is attached. The gear ring drives the brake through a pinion with a gear ratio of 8:1. For the tape and rope designs a drum or pulley was used, whereas the wire design has a clock type power spring fitted inside the drum to enable the unit to fully rewind the lifeline back onto the drum (figures 6.1, 6.2 and 6.3).

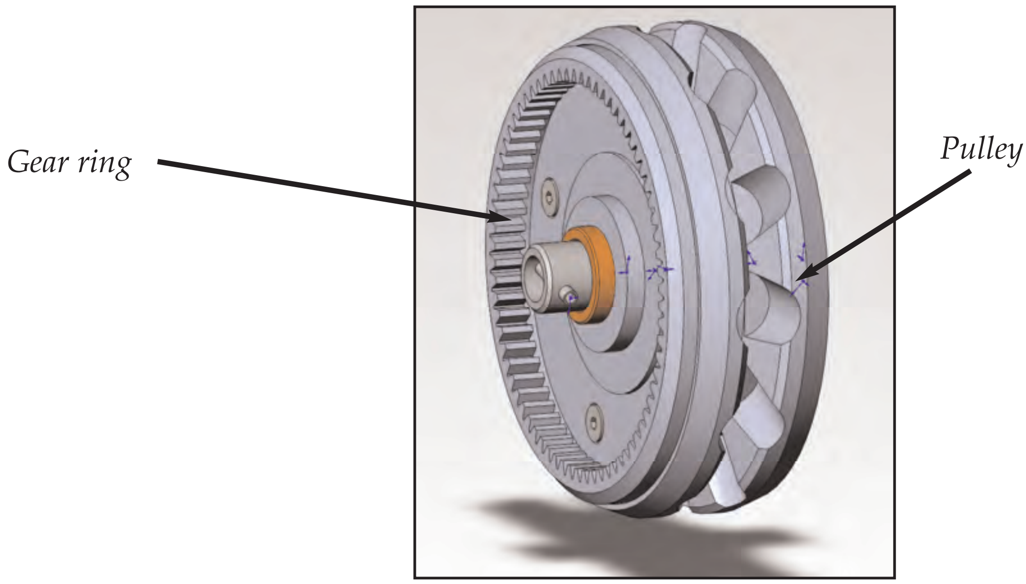


figure 6.1 : pulley drum drive set up for fibre rope

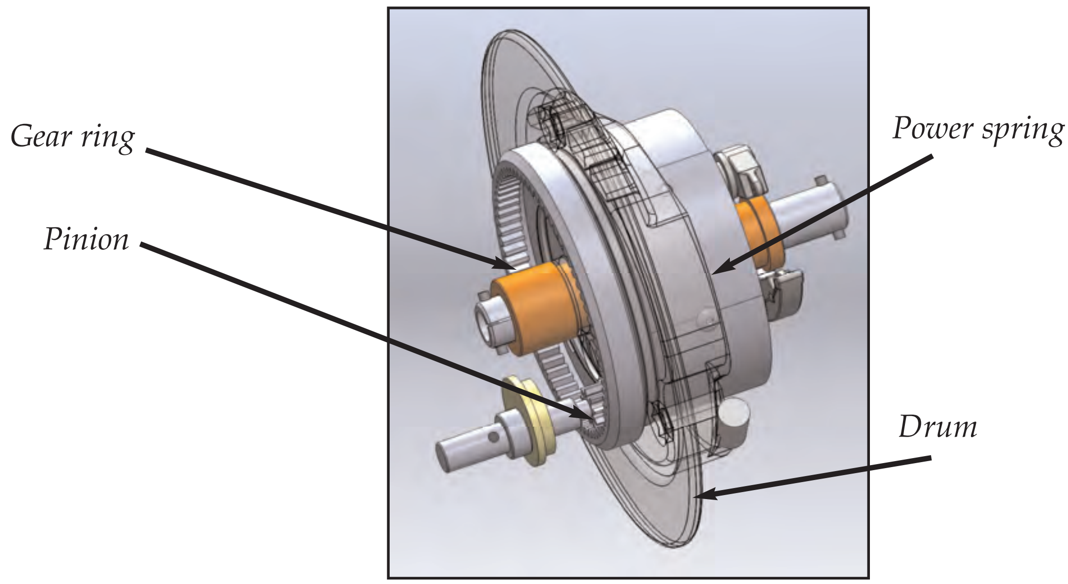


figure 6.2 : partial drum shown with gear ring and integral power spring

The pinion drives the brake hub with the brake shoes rotating within a brake ring to control the descent. These are all contained in a housing. The fibre rope would be contained external to the housing as the diameter of the rope is 9 mm and would make housing it internally impractical.The lack of rewind means that the unit can have a reciprocating action with braking in both directions. As one person descends the other end of the lifeline goes up enabling a second person to descend using the other end. This would enable mass evacuation as with the rewind arrangement, but from the higher levels.

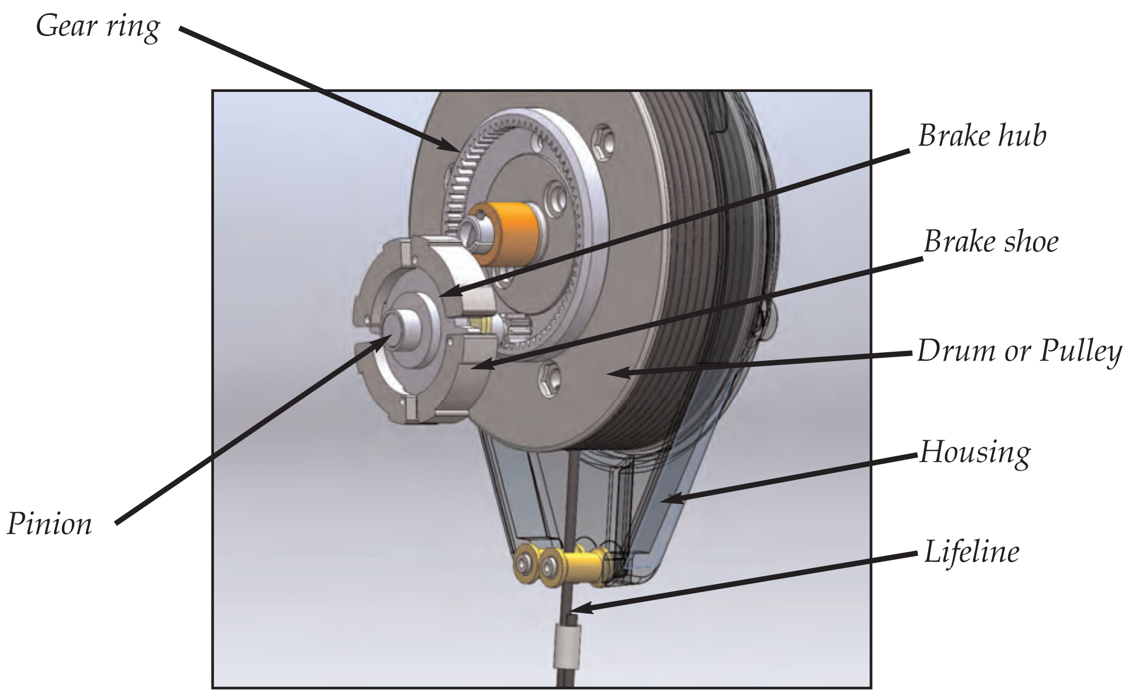


figure 6.3 : configuration showing the drive gear to the brake hub.

6.3 Brake design

The study continued with an investigation of potential brake configurations, where the design aim was to meet the specifications as detailed by the International standards. In the first instance the brake must contain the descent speed to lower than 2 m/s, but must meet the energy rating which is based on a number of drops using a 75 kg weight. There are currently four classifications for descenders the namely:

Class A:	descent energy W >	$7.5\ 10^6$	(J)
Class B:	descent energy W >	$1.5\ 10^6$	(J)
Class C:	descent energy W >	$0.5\ 10^6$	(J)
Class D:	descent energy W >	$0.02\ 10^6$	(J)

Descent energy $W = m \times g \times h \times n$

W = Descent energy, $Joule$

m = test mass, kg

g = gravity, $9.81\ m/s^2$

h = decent height, m

n = number of descents

The test mass is 75 kg and the outside temperature of the unit must not exceed 48^{0}C at any point that an individual could touch

6.3.1 Four sliding brake shoe arrangement

Each brake shoe is able to slide out independently of each other. The configuration shown in figure 6.4 has four brake shoes that slide out from a central hub and engage on a brake band made of steel. The housing is made from 50% glass reinforced nylon following several trials with polymers, Grivory(appendix 3) was chosen as it retained its mechanical properties at elevated temperatures with high mechanical strength, this material has been used for all the designs.
The brake shoes are cast stainless steel and the hub is high tensile strength brass. The shoes have a friction material (D3701,D3910 and D3921 appendix 3) which is common to all designs and follows the running tests over several years of this research. All friction materials gave almost identical results, figures 6.5 and 6.6 show further brake detail.

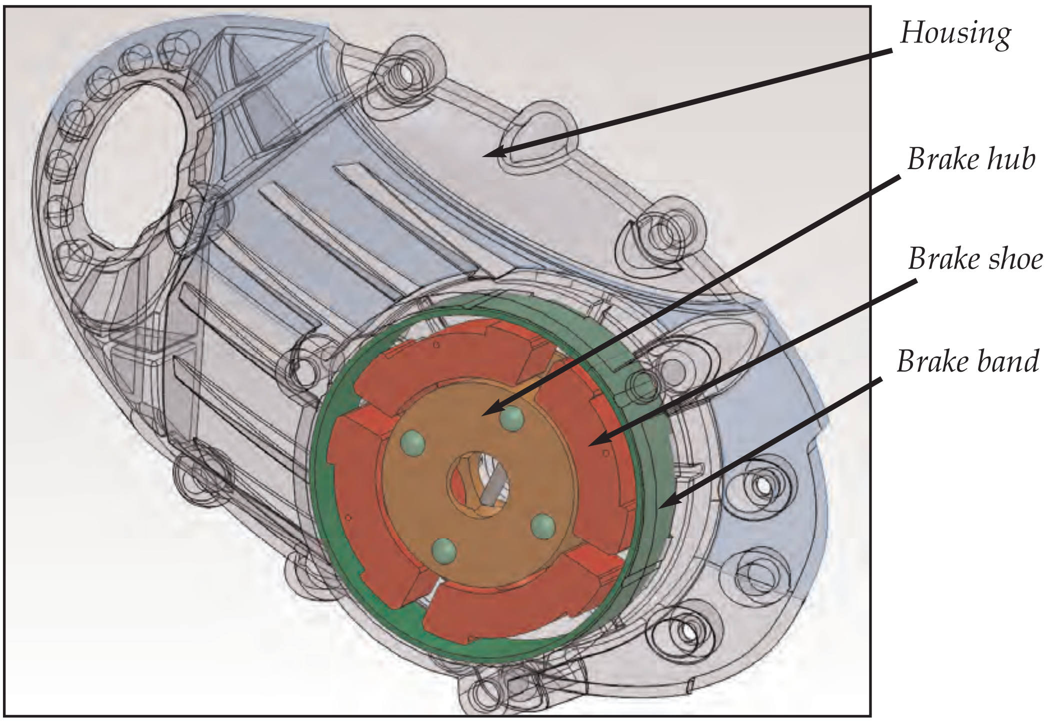


figure 6.4 : end view of sliding brake configuration

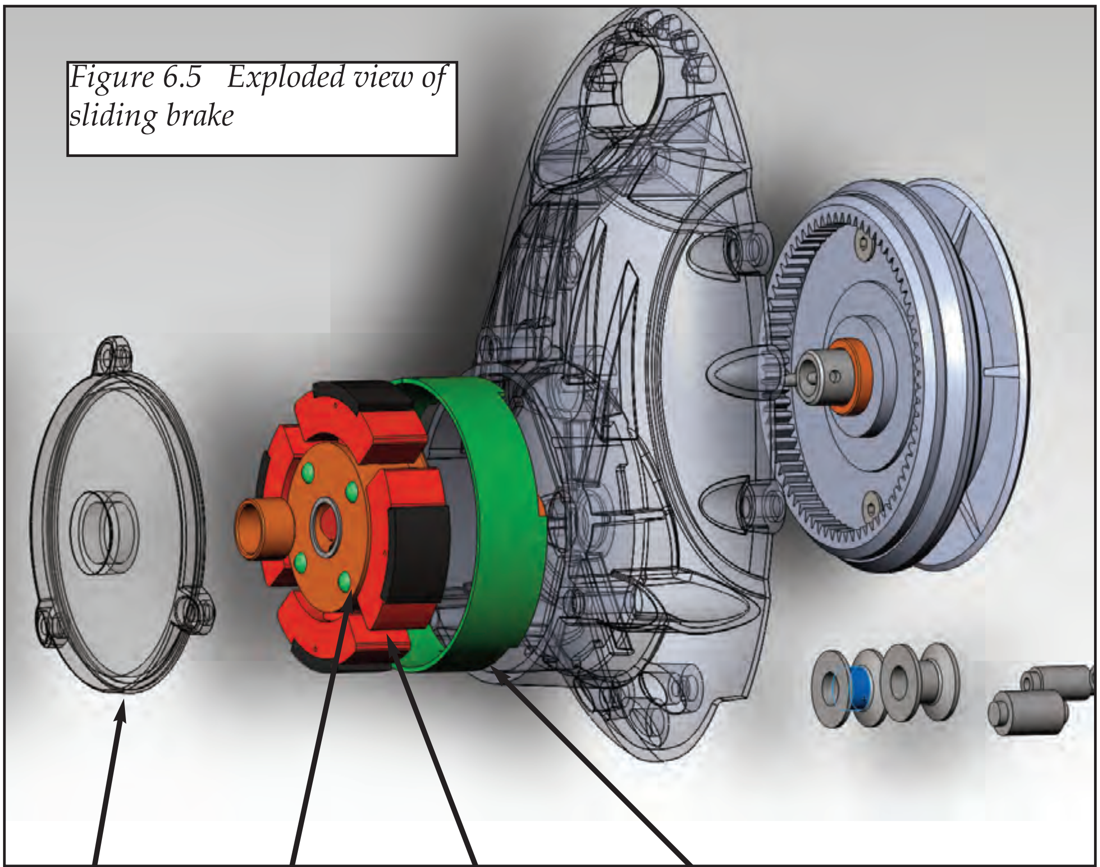


Figure 6.5 Exploded view of sliding brake

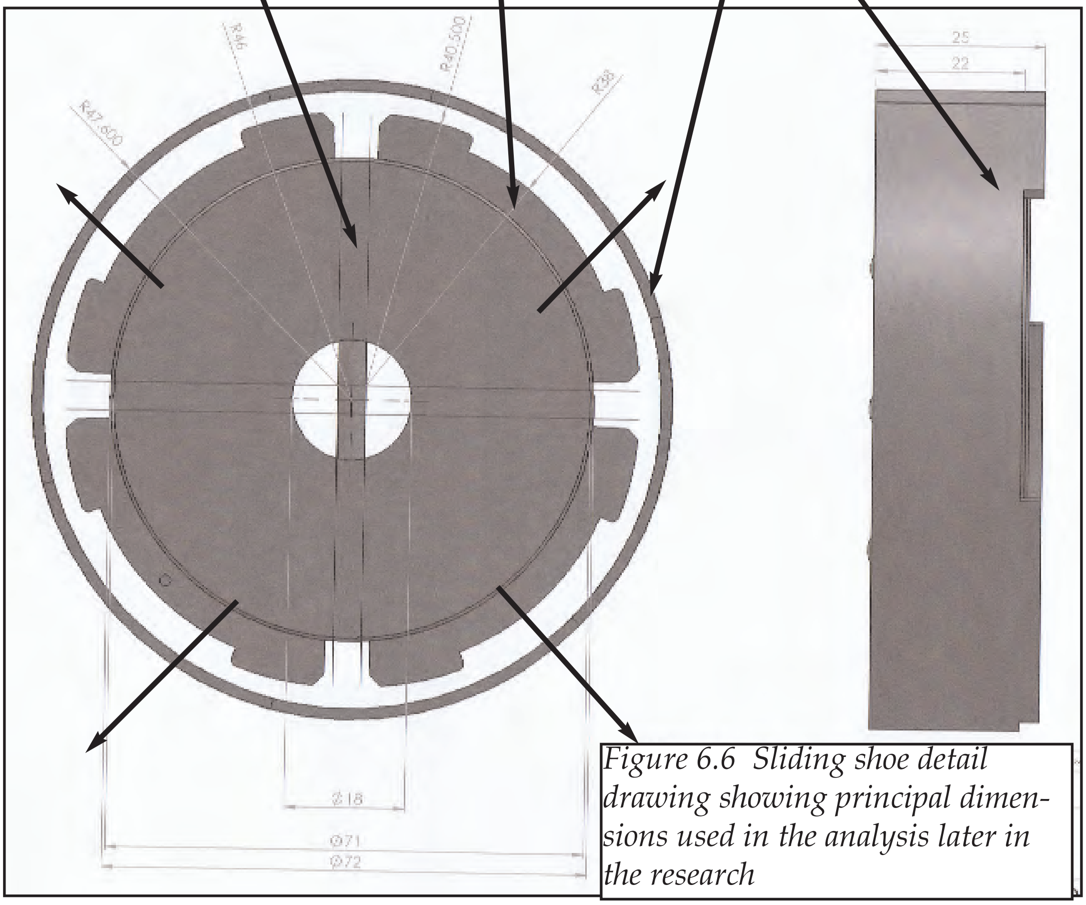


Figure 6.6 Sliding shoe detail drawing showing principal dimensions used in the analysis later in the research

6.3.2 Two pivoting brake shoe arrangement - figure 6.7

The study continued with an investigation of a two-brake shoe arrangement. The objective was to have a compact design that would work in particular with rewinding descenders. In this approach the two shoes pivot from the brake hub and engage on the brake band which is contained in the brake housing. All materials used are similar to the other designs. However, in this instance the brake shoe is cast from high tensile brass (HTB3 appendix 3). A large brake band replaces the thin band used in the other designs considered. The brake shoes can be arranged in 2 leading shoe, 2 trailing shoe and 1 leading and 1 trailing shoe configurations. The torque produced by the brake shoes can be provided either in one direction for a rewinding descender or in both directions for a reciprocating descender. The drum which has a gear ring attached drives a pinion that has the brake hub pinned to it as in the other design, figures 6.8 and 6.9 show further design detail for the brake.

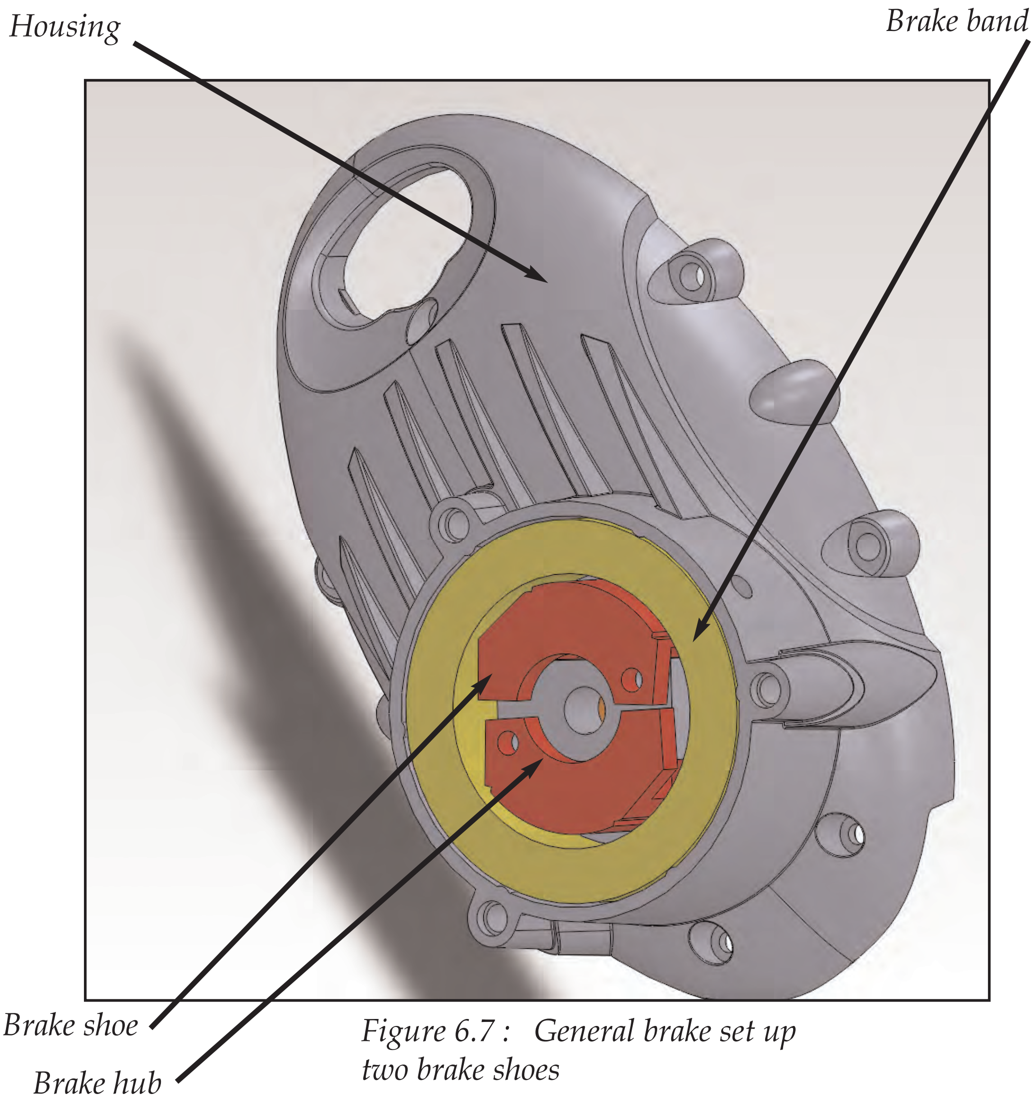


Figure 6.7 : General brake set up two brake shoes

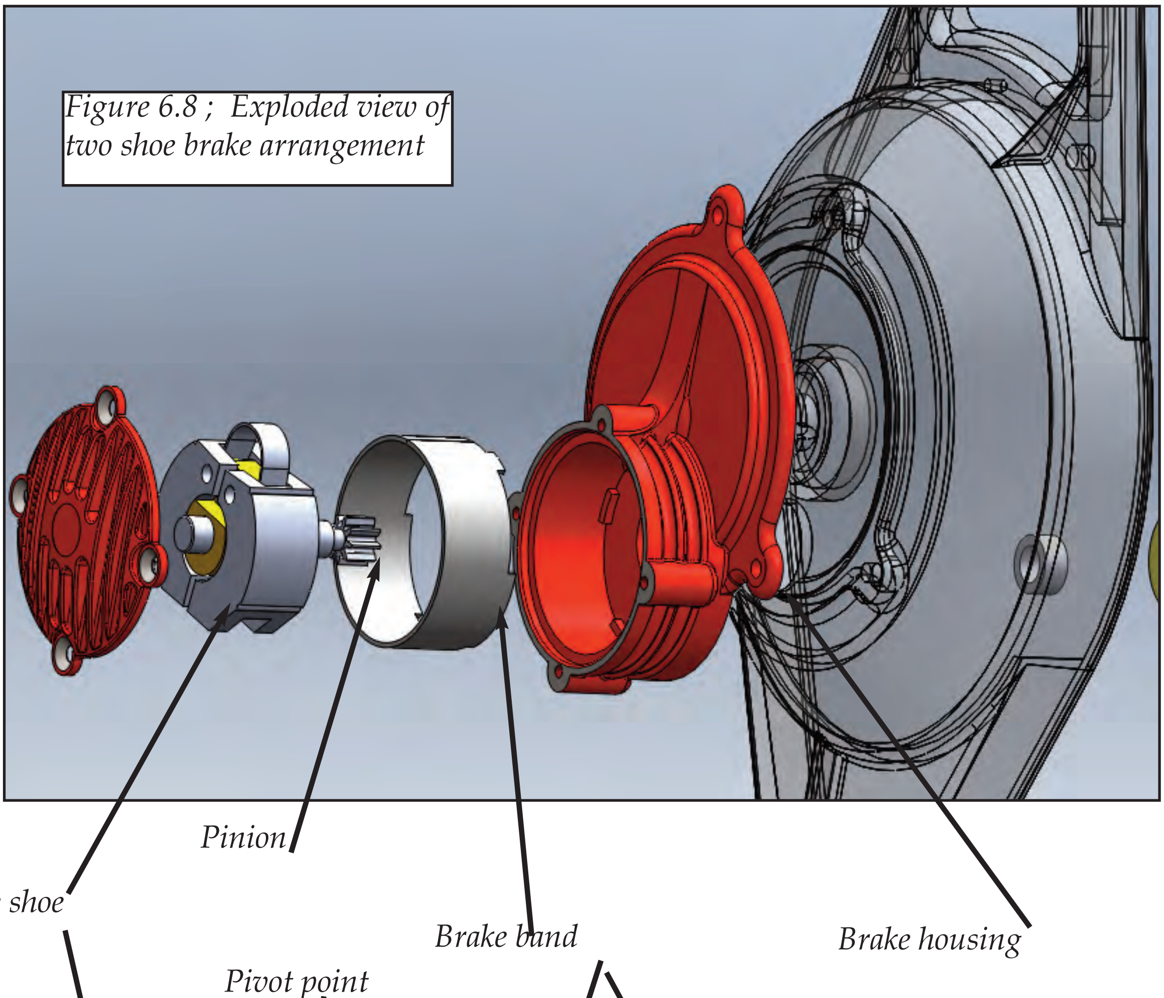


Figure 6.8 ; Exploded view of two shoe brake arrangement

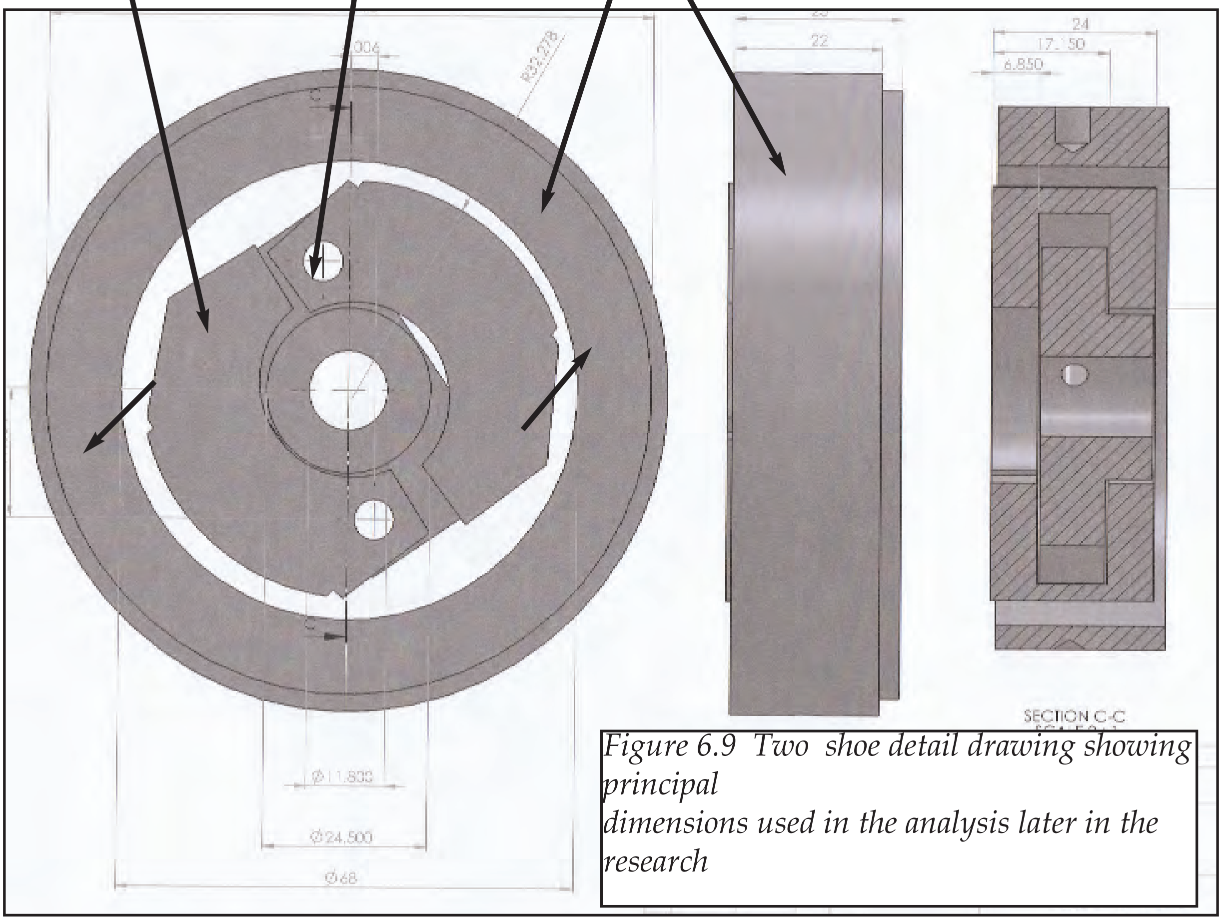


Figure 6.9 Two shoe detail drawing showing principal dimensions used in the analysis later in the research

6.3.3 Two leading, two trailing pivoting brake shoe arrangement - figure 6.10

This configuration has four shoes that are mounted on a brake hub. The brake hub is pinned to the pinion, which in turn is driven by the ring gear mounted on the drum. By having shoes in both leading and trailing it can operate in both directions permitting use in reciprocating designs, as well as rewind types. However, with regard to rewind operation the two shoes will create braking that the rewind power spring would have to overcome. The brake shoe is based upon the sliding brake shoe with sliding spigot removed and a pivot point included. The shoe had to be machined on the reverse side in order to allow for unrestricted movement and closure when not engaged on the hub.

Figures 6.11 and 6.12 show the arrangement of the brakes in exploded and detail format.

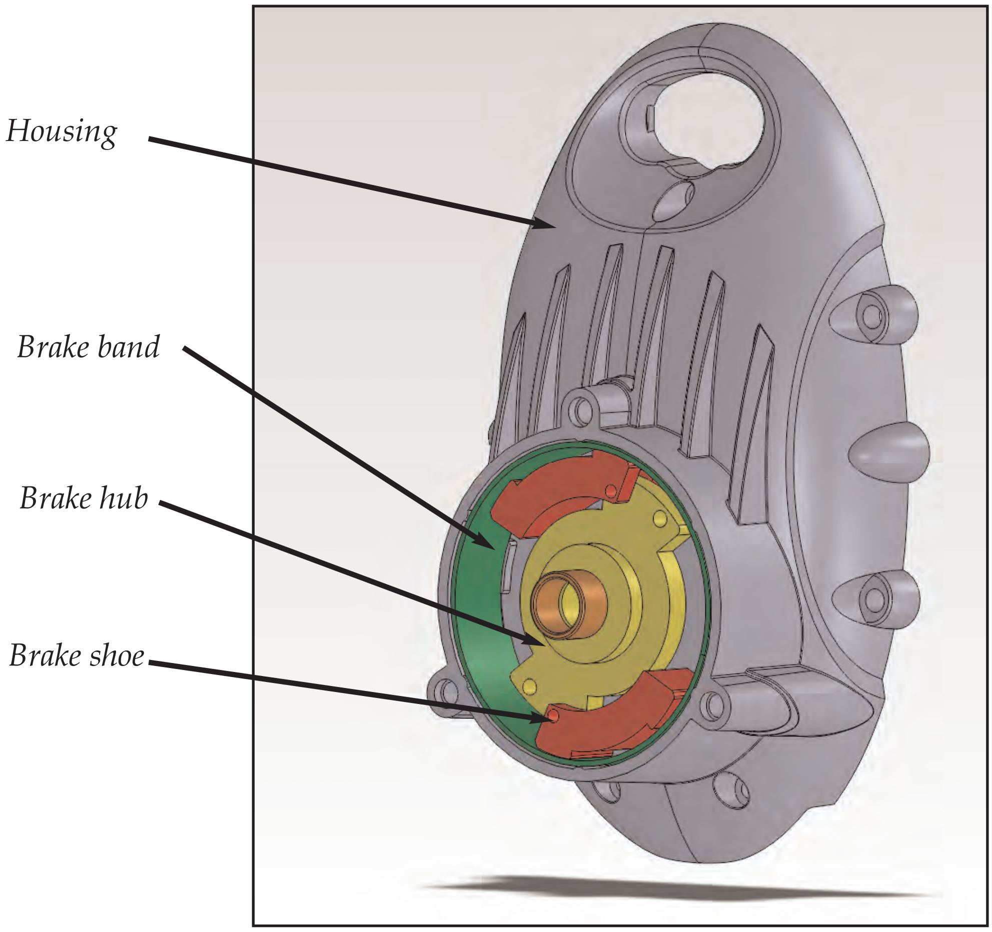


Figure 6.10 : General brake set up two leading and two trailing shoes

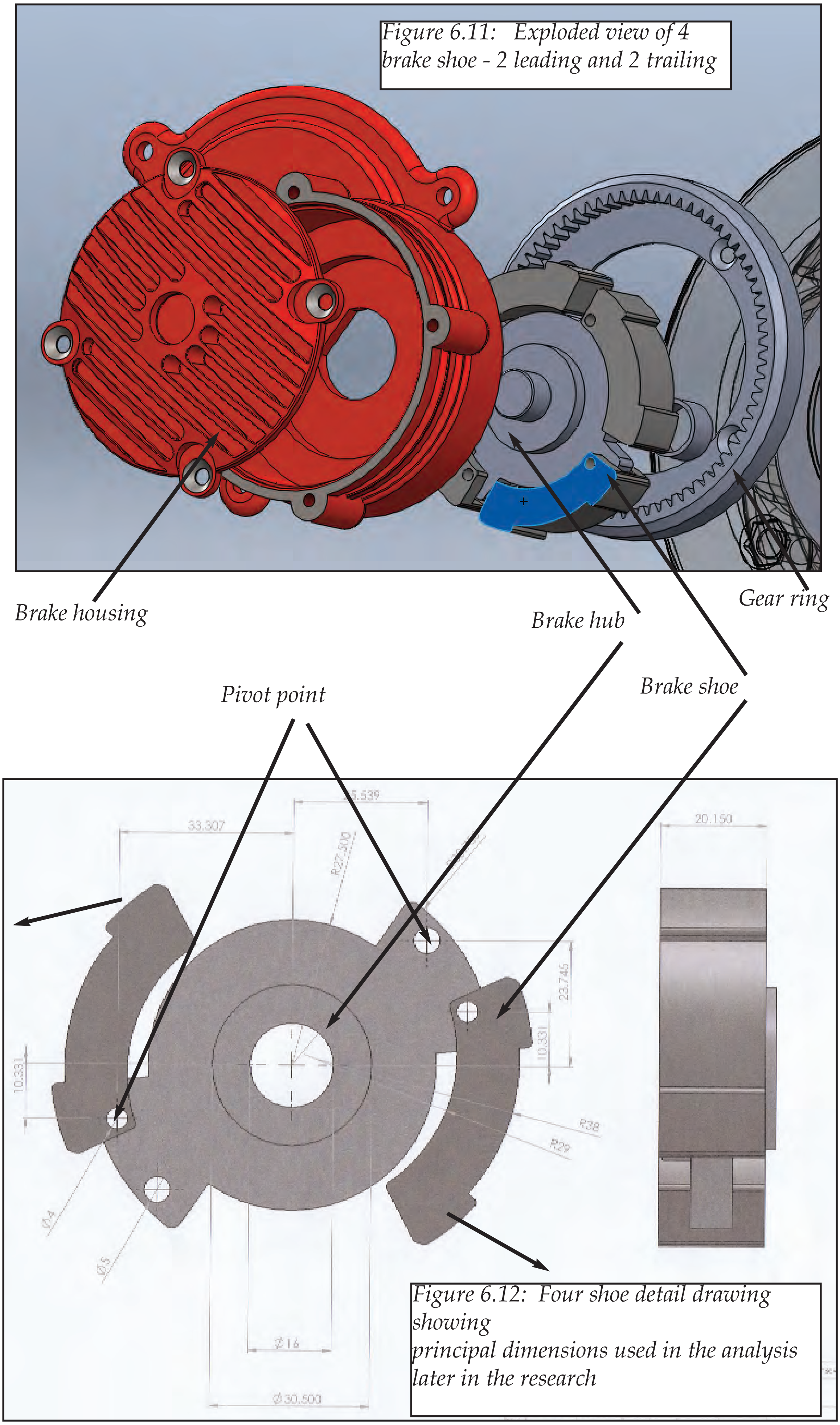


Figure 6.11: Exploded view of 4 brake shoe - 2 leading and 2 trailing

Figure 6.12: Four shoe detail drawing showing
principal dimensions used in the analysis later in the research

6.3.4 Four leading or four trailing brake shoe arrangement - figure 6.13

This configuration has four shoes pivoting on the brake hub and can be configured either as four leading or four trailing options, the brake hub is pinned to the pinion which is driven by a ring gear on the drum as with previous designs. Although it can be configured as four trailing shoe its normal configuration is four leading shoes. During the study this was investigated as it is anticipated that it would provide the best solution for rewinding descenders but not suitable for reciprocating designs. figures 6.14 and 6.15 show the exploded view of the braking arrangement together with detail drawing

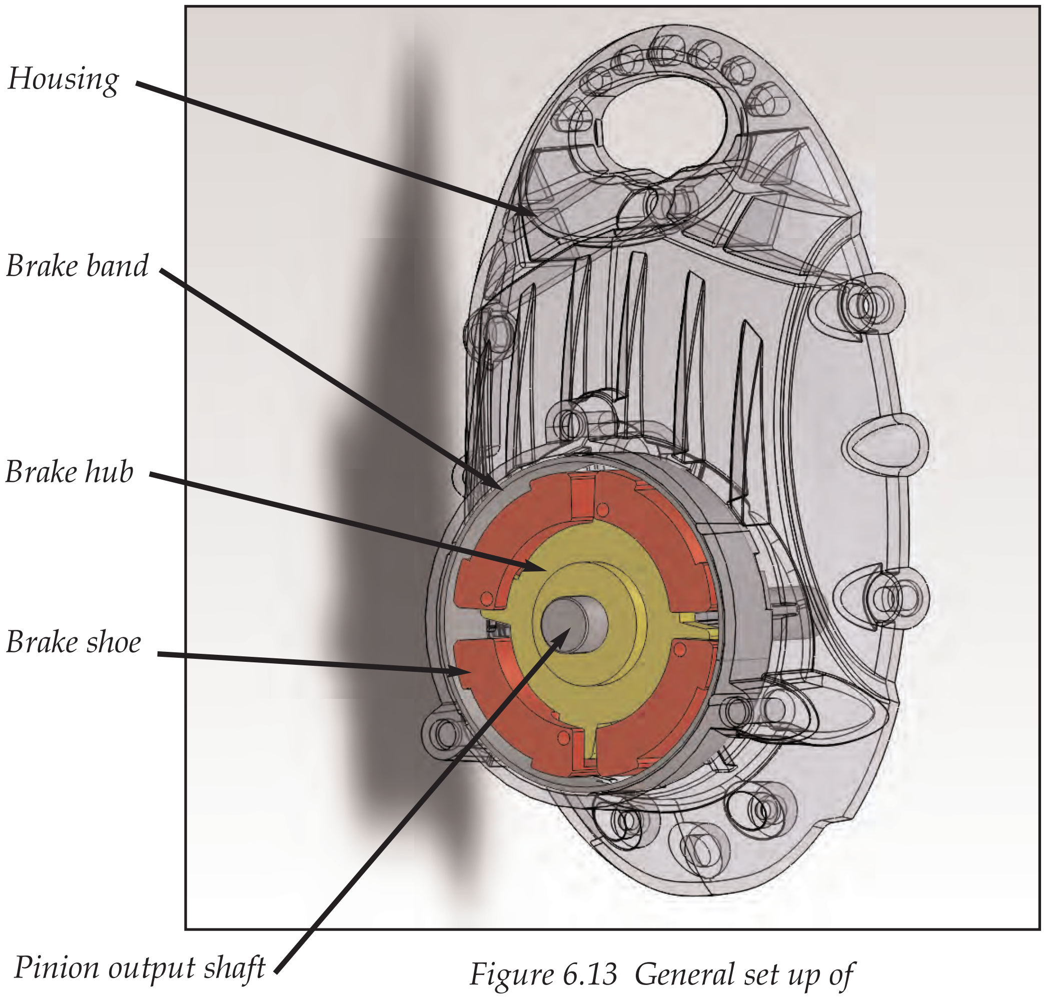


Figure 6.13 General set up of four brake shoe design

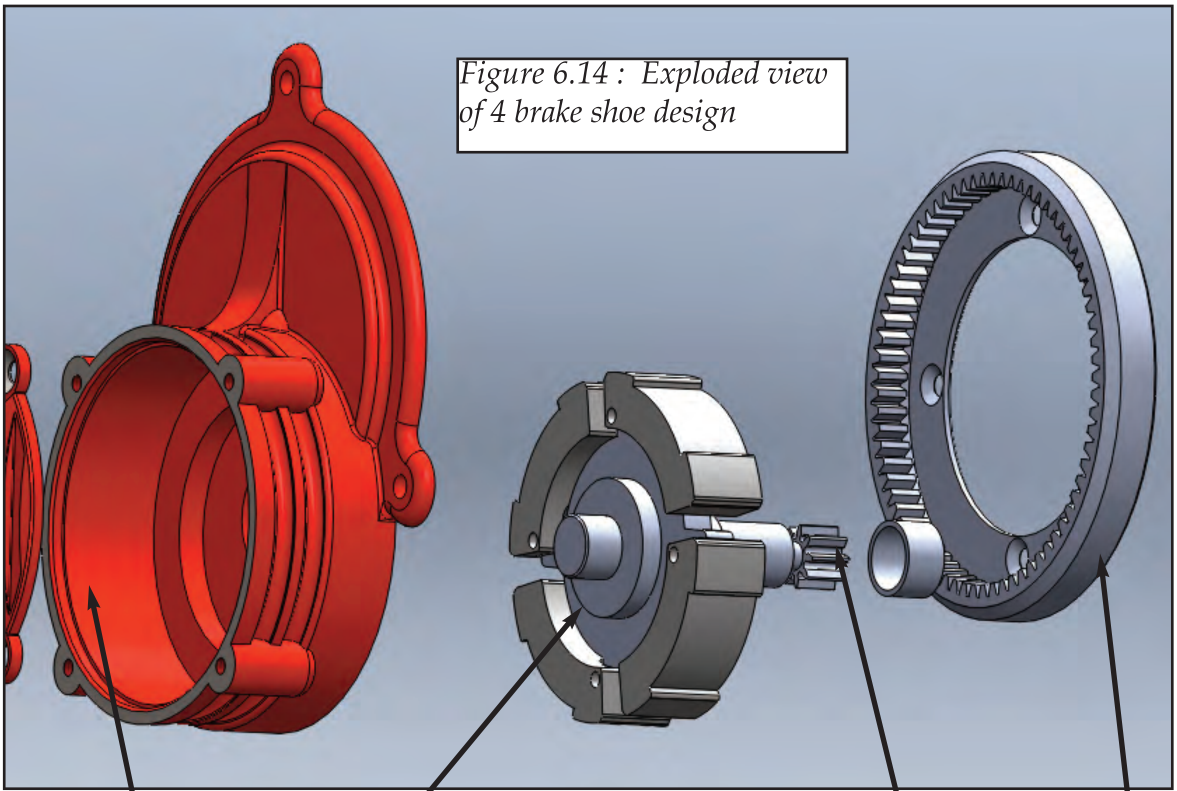


Figure 6.14 : Exploded view of 4 brake shoe design

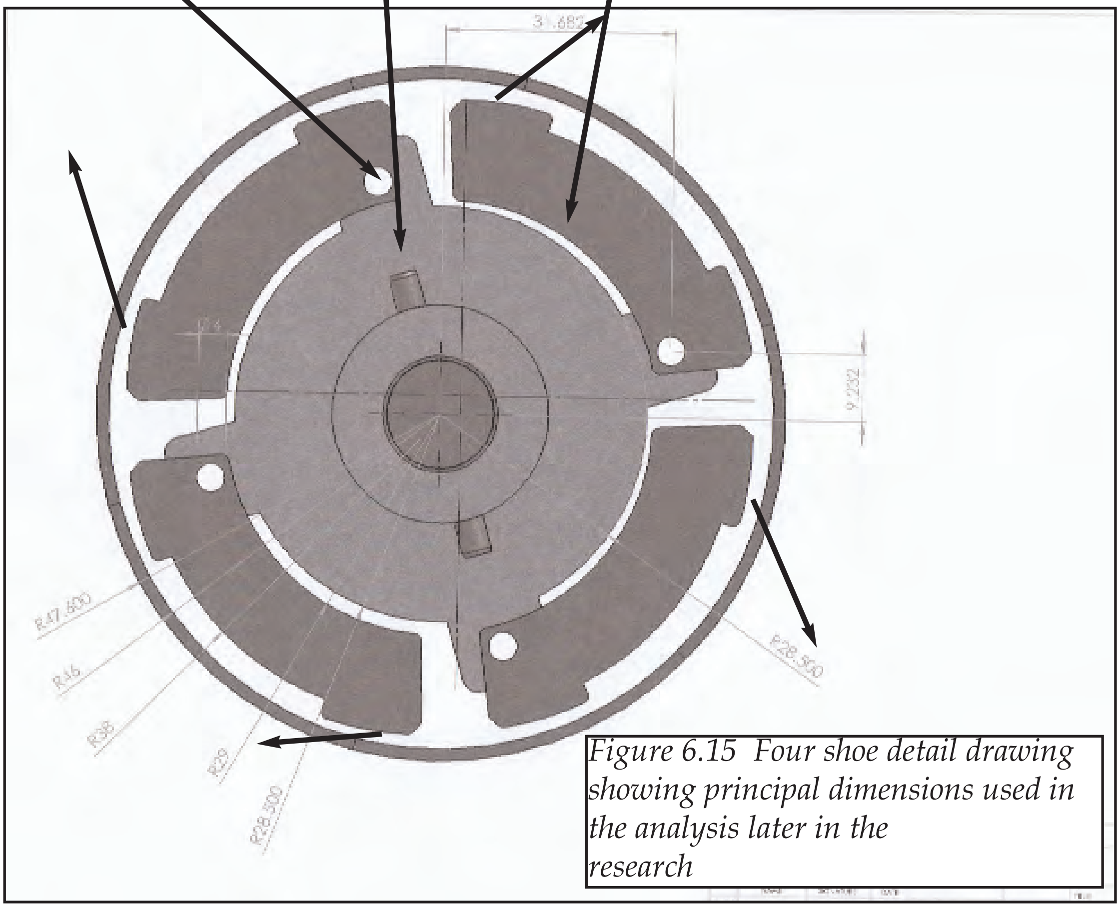


Figure 6.15 Four shoe detail drawing showing principal dimensions used in the analysis later in the research

6.3.5 Fibre rope continuous over pulley design

Figure 6.16 shows the descender design with four sliding brake shoes, it is shown with two brake housing cavities but only one is being used. The diagram shows the design with a transparent window on the housing, so that the internal arrangement is visible. The rope drives the pulley by a grooved track in the drum. The rope simply passes around the drum from the entry point to the exit point at the base where the entry guide rods are located.
Figure 6.17 shows the brake arrangement and location in figure 6.16, whereas, figure 6.18 shows the drive pulley with gear ring, the drive V arrangement on the pulley is clearly visible.

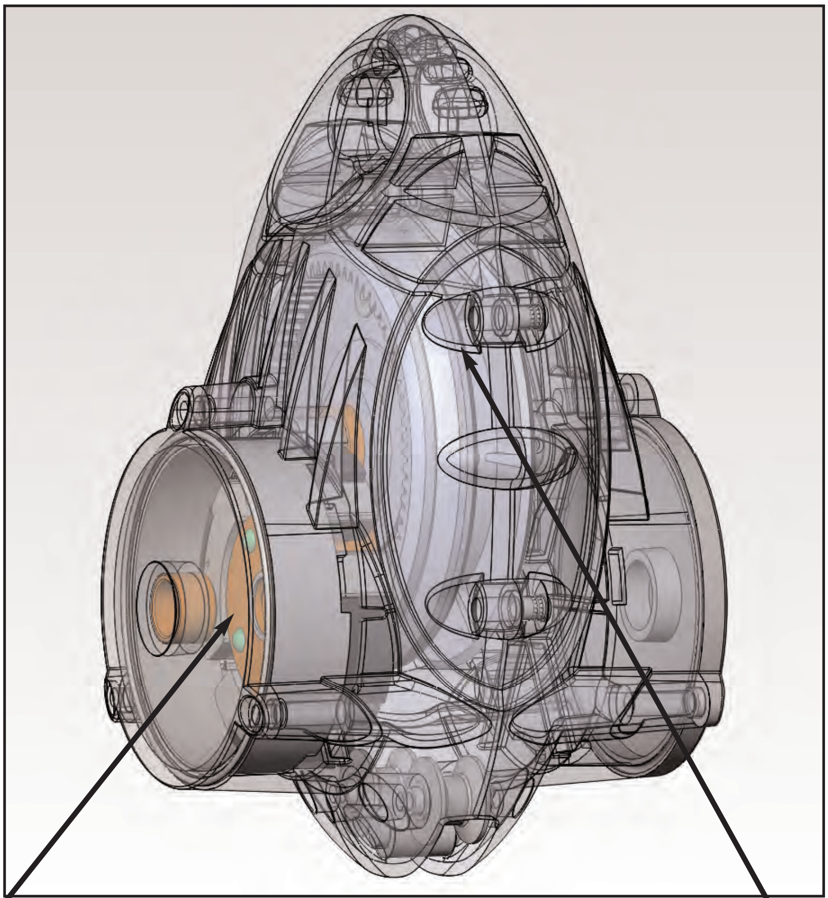


Figure 6.16 Reciprocating design

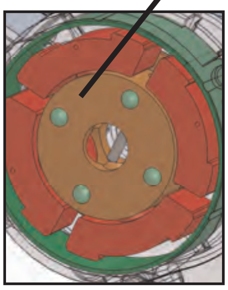


figure 6.17 : 4 sliding brake shoes

figure 6.18 : Geared pulley

6.3.6 Wire rope rewinding design on drum

The configuration has four leading shoes that pivot on the brake hub and that engage on the brake band made of steel. The wire rope passes over the entry guide rods and coils onto the drum under constant tension from the power spring. The descent speed is designed to be 2m/s, but if the drum diameter was increased and the gearing maintained then as with the tape, the torque would be greater for the larger diameter and then reduce as the wire pays out which would give a high initial velocity and then decelerate the person as the reel approaches the design limits of the 15 m unit. Realistically the governing factor would be the power spring which would be a substantial size and weight, this would limit units that recoil to below 50 m (figure 6.19).

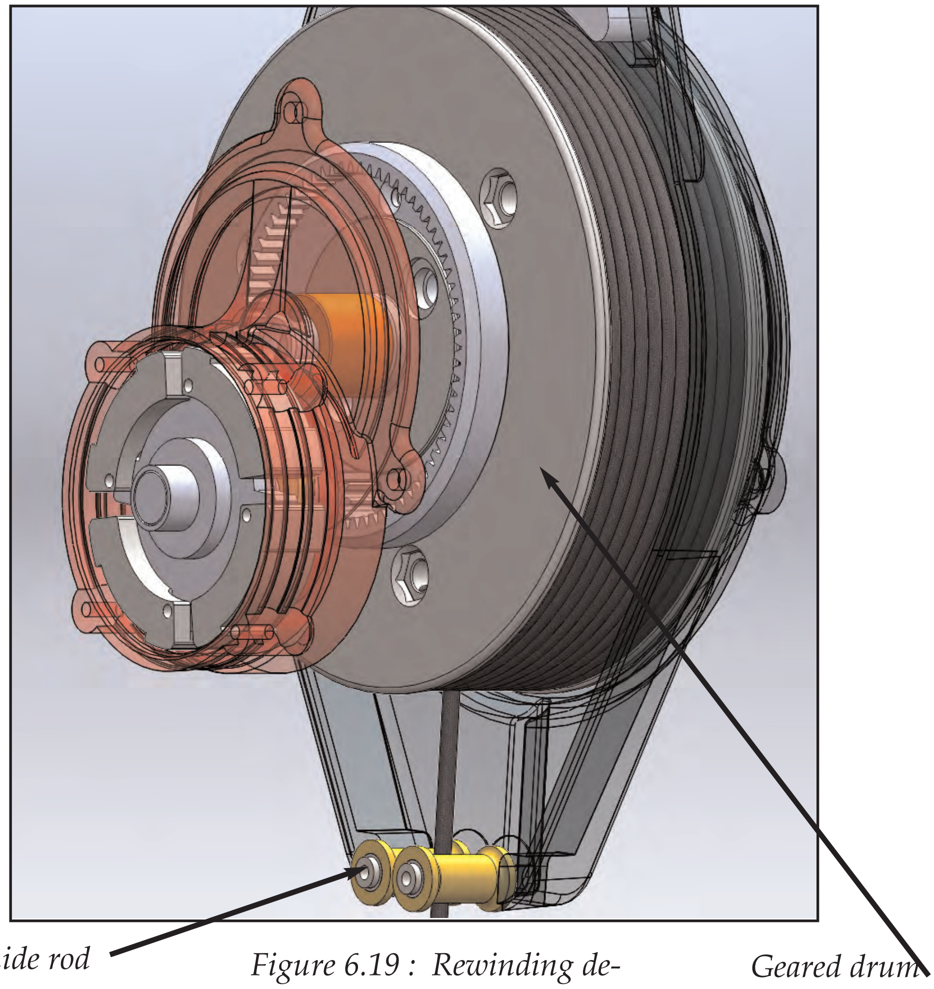


Figure 6.19 : Rewinding design

6.3.7 Tape contained in a spool design

This configuration has four shoes that slide out from a central hub and engage on a brake ring made of steel. The tape is wound on a drum inside a housing and passes out through the exit opening over the entry guide rods.The objective is to have cassettes with different load capacities. The tape would reduce its diameter allowing a faster descent in the initial stage and as the tape pays out the radius would decrease and the torque would decrease slowing the descent. The research considers this reduction in tape radius has a major contributing factor in evacuation as the descent speed can be quite high initially, but can be controlled to an acceptable speed before landing (figure 6.20).

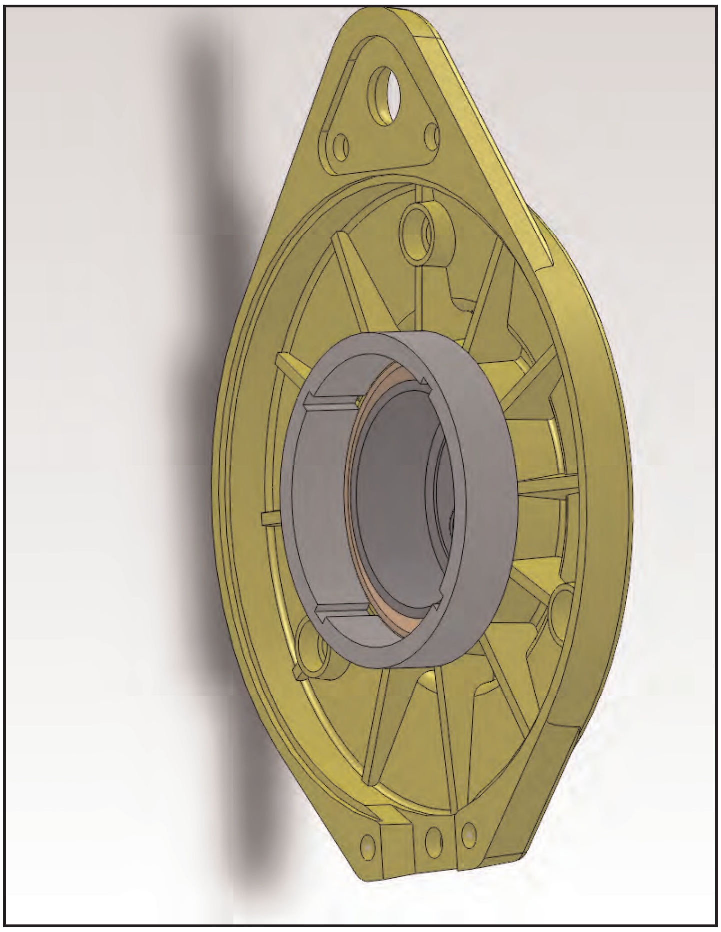


Figure 6.20 Tape design

6.4 Prototypes

6.4.1 The sliding brake shoe concept

A prototype was produced with the four sliding brake shoes as shown in figure 6.22.
This was then fitted into the main prototype for testing figures 6.21 , 6.23 and 6.24 and also in a brake only version using the reciprocating housing for use in the brake test rig. The brake band simply drops into position to allow the band to be easily changed.The brake is supported in bearings located in an end-cap and in the housing. The nature of the prototype construction meant that the individual brake set ups could be simply changedbetween successive tests.

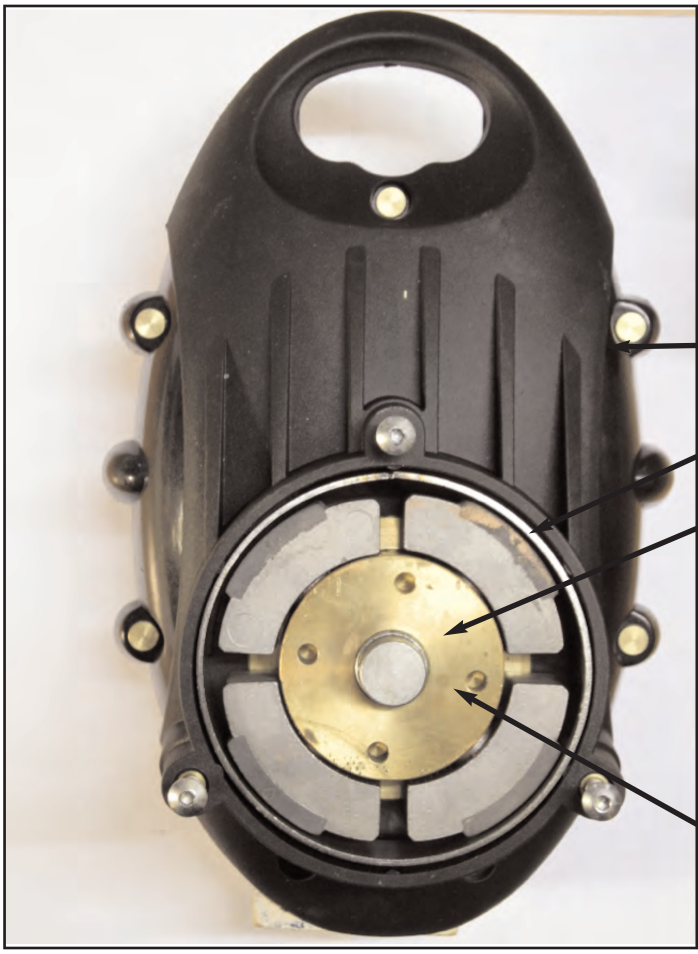


Figure 6.21: Reciprocating design concept - sliding shoe

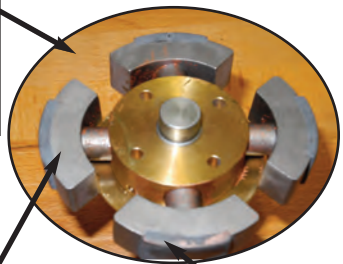


Figure 6.22: Exploded view sliding brake

Two models were prototyped in order to evaluate all aspects of the design; both static and dynamic. One a reciprocating design and the other a self-recoiling design are shown in the two photographs figures 6.23 and 6.24.
The models were such that all the brake designs were able to be fitted into the prototypes in order to evaluate the advantages and disadvantages of each design. The housings were injection moulded in Grivory (appendix 3) and they were able to be evaluated in the brake torque test rig (section 8.5). By approaching the testing in this manner not only could the theory be validated, but also the functionality of the brake could be tested. In addition, by removal of the power spring and modifying the drum the rewind prototype was configured to take 60 m of tape for use with the drop test rig to evaluate the performance with varying torque (section 8.4).

Figure 6.23: Rewind design prototype

Figure 6.24: Reciprocating design prototype

6.4.2 Two pivoting shoe brake configuration

The brake was prototyped so that the three variants could be tested, one leading and one trailing brake shoe main figure 6.25 and figure 6.27, two leading brake shoes figure 6.26 and 2 trailing brake shoes.
The brake band diameter was reduced and in order to run in the same units its thickness was increased. The brake shoes were prototype cast in HTB 3 (high tensile brass) in this case the brake hub was made from aluminium. The brake band was made in aluminium which is not a usual brake surface but it has good thermal properties, so the heat dissipation was considered good providing a heat sink in the polymer shell.

Figure 6.25: Reciprocating design concept - two shoe

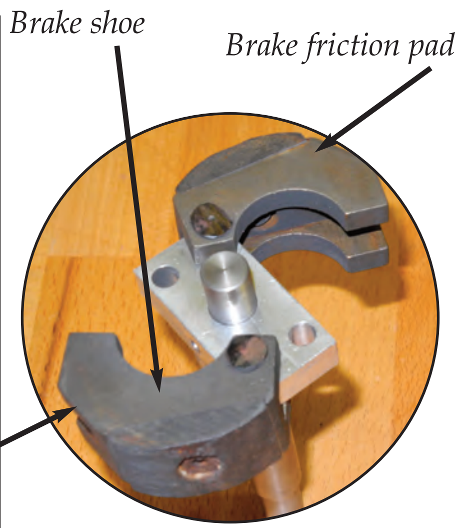


Figure 6.26 : Two leading brake shoe configuration

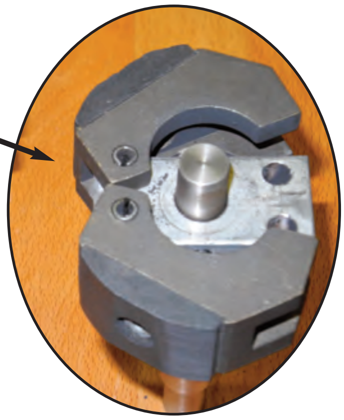


Figure 6.27: One leading and one trailing brake shoe configuration

6.4.3 Two leading, two trailing pivoting brake shoes configuration

With a hub that has pivot points adjacent to each other as shown in figures 6.28 and 6.29 the brake shoes can be mounted on a two leading or two trailing shoe configuration, depending on the direction of rotation. The prototype, as with all the other configurations, can run in static and dynamic tests, in addition to being able to be mounted and run in the brake test rig.

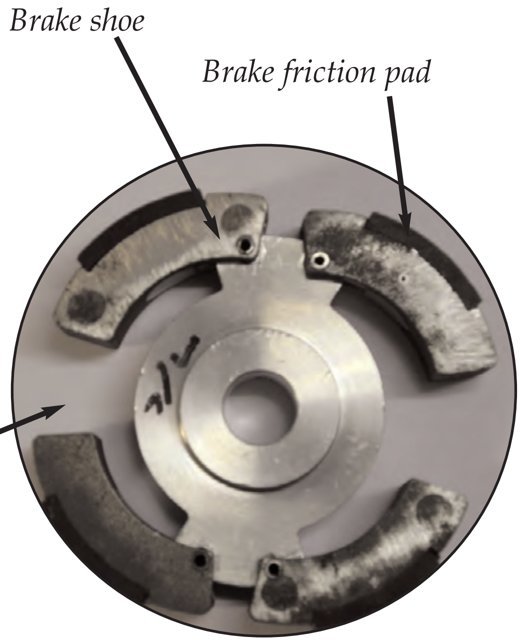


Figure 6.29 Brake shoe exploded to show pivot layout

Figure 6.28: Reciprocating design concept - 4 shoe, combined leading and trailing

6.4.4 Four leading brake shoe concept

This configuration as shown in figure 6.30 has four shoes that pivot on a brake hub, Two brake models were produced and the pivot point was moved to investigate how the position of the pivot influences its performance as shown in figures 6.31 and 6.32. The brakes were configured in a four leading or four trailing configuration, depending on the direction of rotation.

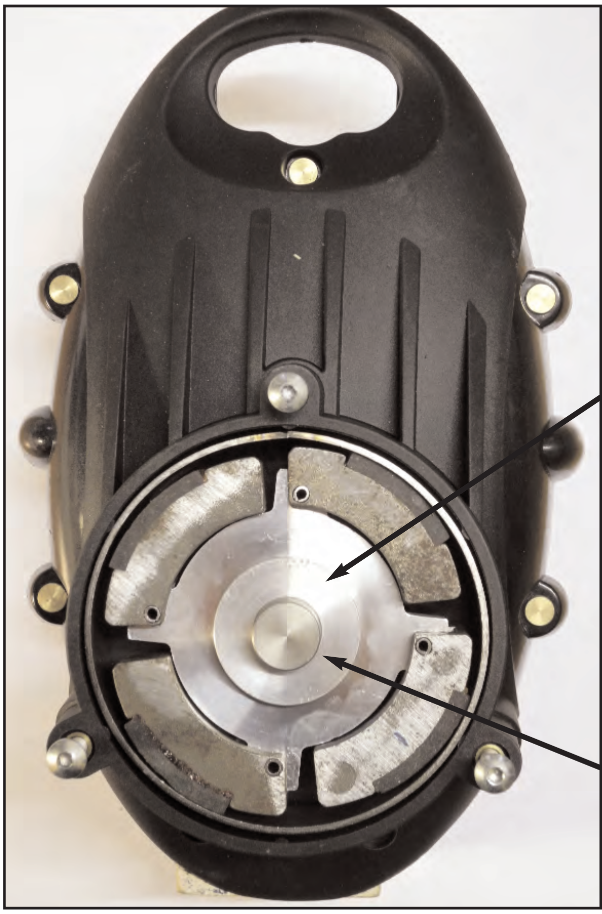


Figure 6.30: Reciprocating design concept - 4 shoe leading and trailing

Figure 6.31 : Pivot point A2 inboard

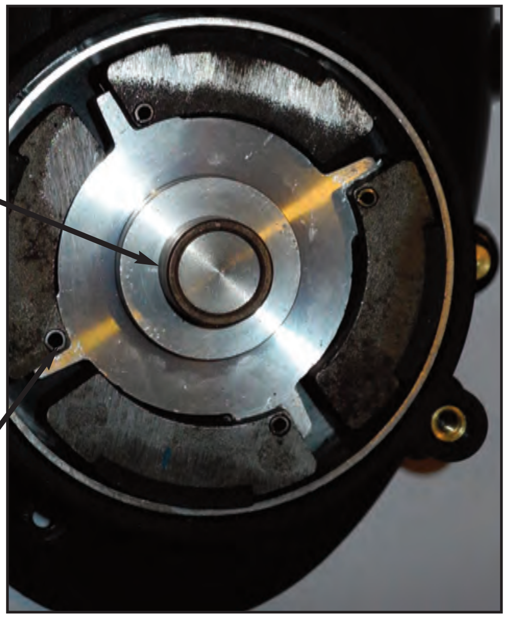


Figure 6.32: Pivot point A1 outboard

6.5 Discussion and summary

In this chapter a number of configurations with different brake shoe arrangements have been described and developed.
These can be either rewinding or not. Following on from the design, construction prototypes were manufactured and tested using the test rig (chapter 8) and in an actual field test (chapter 8). The casings were manufactured using prototype plastic injection mould tooling and the parts moulded in Grivory (appendix 3), which was chosen due to its good mechanical properties, Grivory which is glass filled is a high strength high impact polymer that retains its mechanical properties at elevated temperatures. The three shoe types were developed to enable direct comparison with each other in respect of the two configurations of four shoe brakes and also in comparison with the two shoe brake. The brake friction materials were directly bonded to the brake shoes to remove a possible variable. The designs shown were compact with consideration to the other aims such as weight and portability.

Chapter 7: Analysis of the designs

7.1 Introduction

In this chapter an analysis is made of the braking mechanisms to develop a theory which is validated in a subsequent chapter. There are three main designs considered, starting with a 4 shoe sliding brake and then proceeding with 2 and 4 shoe pivot brakes, with leading and trailing shoes.

The analysis provide equations which relate brake torques to brake speed for all the brake shoe arrangements and the graphs obtained from these equations are compared with test results in the following chapter.

More analysis was carried out to provide equations which relate drop velocity to drop length, when using tape on a spool. These result in a high initial velocity, when the spool is full and a low final velocity, when the spool is empty. However, at the point that the evacuee makes ground contact then the evacuee is required to be moving at less than 2 m/s.

7.2 Analysis of sliding brake shoe

The configuration comprises 4 brake shoes that slide along a guide within a brake hub driven by a ring and pinion gear chain with a ratio of 8.

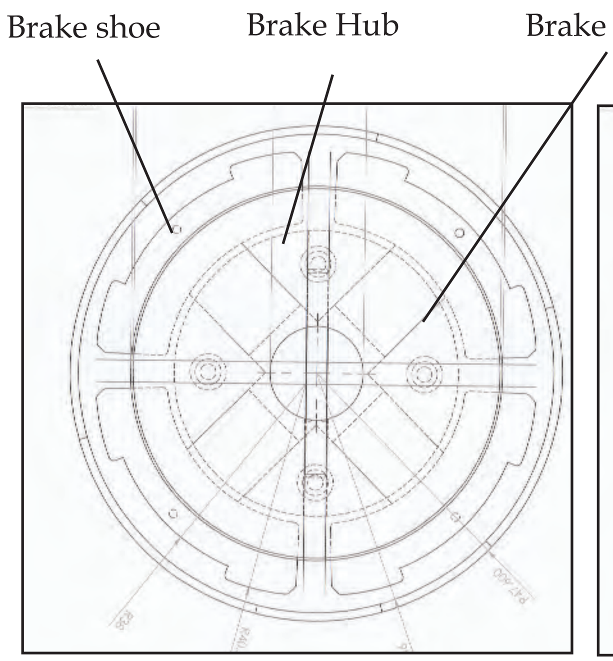


figure 7.1: Schematic sliding brake

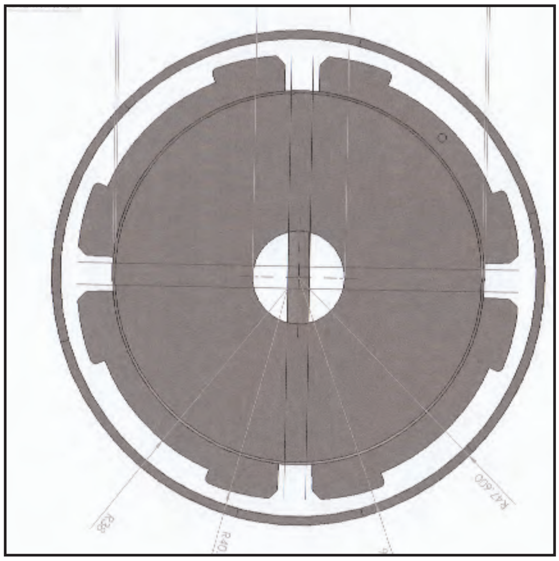


figure 7.2 : Negative view sliding brake

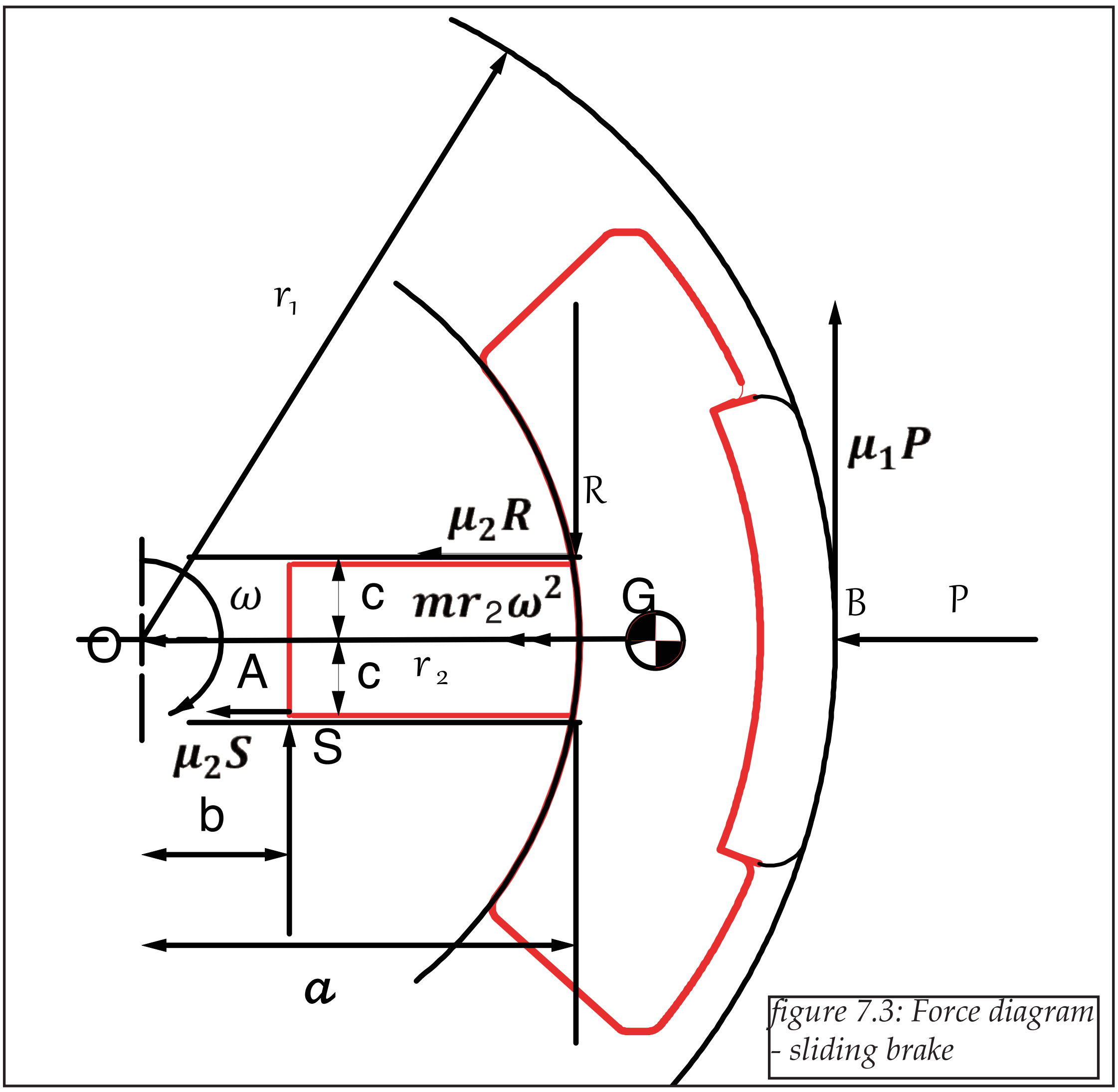


figure 7.3: Force diagram - sliding brake

The above *figure 7.3* is a schematic drawing of the concept of sliding brake drawn to a scale of 2:1. The centre of mass of the shoe is at G and the centre of the brake band is at 0. The following points are also shown:

r_1	Brake band radius = 0.046 m
r_2	Radial distance from G to 0 = 0.034 m
$\mu_1 P$	Tangential force representing the frictional force on the shoe
$mr_2\omega^2$	The radial mass acceleration effective force at G
μ_1	The coefficient of friction for the brake material
μ_2	The coefficient of friction for the sliding element

Taking moments

$$\sum_H \quad mr_2\omega^2 = P + \mu_2 R + \mu_2 S \ldots\ldots\ldots\ldots\ldots \text{(7.2.1)}$$

$$\sum_V \quad 0 = \mu_1 P + S - R \ldots\ldots\ldots\ldots\ldots \text{(7.2.2)}$$

$$\sum_{mA} \quad mr_2\omega^2\, c = Pc + \mu_1 P(r_1 - b) + \mu_2 R2c - R(a - b) \ldots \text{(7.2.3)}$$

$$\therefore \; R(a - b - 2\mu_2 c) = P(\mu_1 r_1 - \mu_1 b + c) - mr_2\omega^2 c \ldots\ldots\ldots \text{7.2.3a)}$$

From equation 7.2.1, $$P = mr_2\omega^2 - \mu_2(R + S) \ldots\ldots\ldots\ldots \text{(7.2.4)}$$

From equation 7.2.2 and 7.2.4, $$P = mr_2\omega^2 - \mu_2(R + R - \mu_1 P) \ldots\ldots \text{(7.2.5)}$$

$$\therefore \; P(1 - \mu_1\mu_2) = mr_2\omega^2 - 2\mu_2 R \ldots\ldots\ldots\ldots\ldots \text{(7.2.5a)}$$

From 7.2.3a and 7.2.5a,

$$P(1 - \mu_1\mu_2) = mr_2\omega^2 - 20\mu_2\left(\frac{P(\mu_1 r_1 - \mu_1 b + c) - mr_2\omega^2 c}{a - b - 2\mu_2 c}\right) \ldots \text{(7.2.6)}$$

$$\therefore \; P\left(1 - \mu_1\mu_2 + 2\mu_2\left(\frac{\mu_1 r_1 - \mu_1 b + c}{a - b - 2\mu_2 c}\right)\right) = mr_2\omega^2\left(1 + \frac{2\mu_2 c}{a - b - 2\mu_2 c}\right) \text{(7.2.7)}$$

Brake torque per shoe = $P\mu_1 r_1$ (7.2.8)

Assume that μ_2 is small from equations 7.2.6 and 7.2.7 the brake torque for four shoes in terms of ω^2 can be evaluated

Brake torque = $4\mu r_1 r_2 \omega^2 = 16\mu m r_1 r_2 \pi^2 N^2$ ……..(7.2.9)

since $2\pi N = \omega$……………...(7.2.9a)

Although there is bound to be some stiction as a result of the brake shoe engaging with the brake band by moving along the slides, when in operation the vibration set up by the imperfections in the brake band and other small variations in frictional force is sufficient in itself to eliminate most of the effects of stiction. Therefore, the analysis can be simplified as drawn in figure 7.4

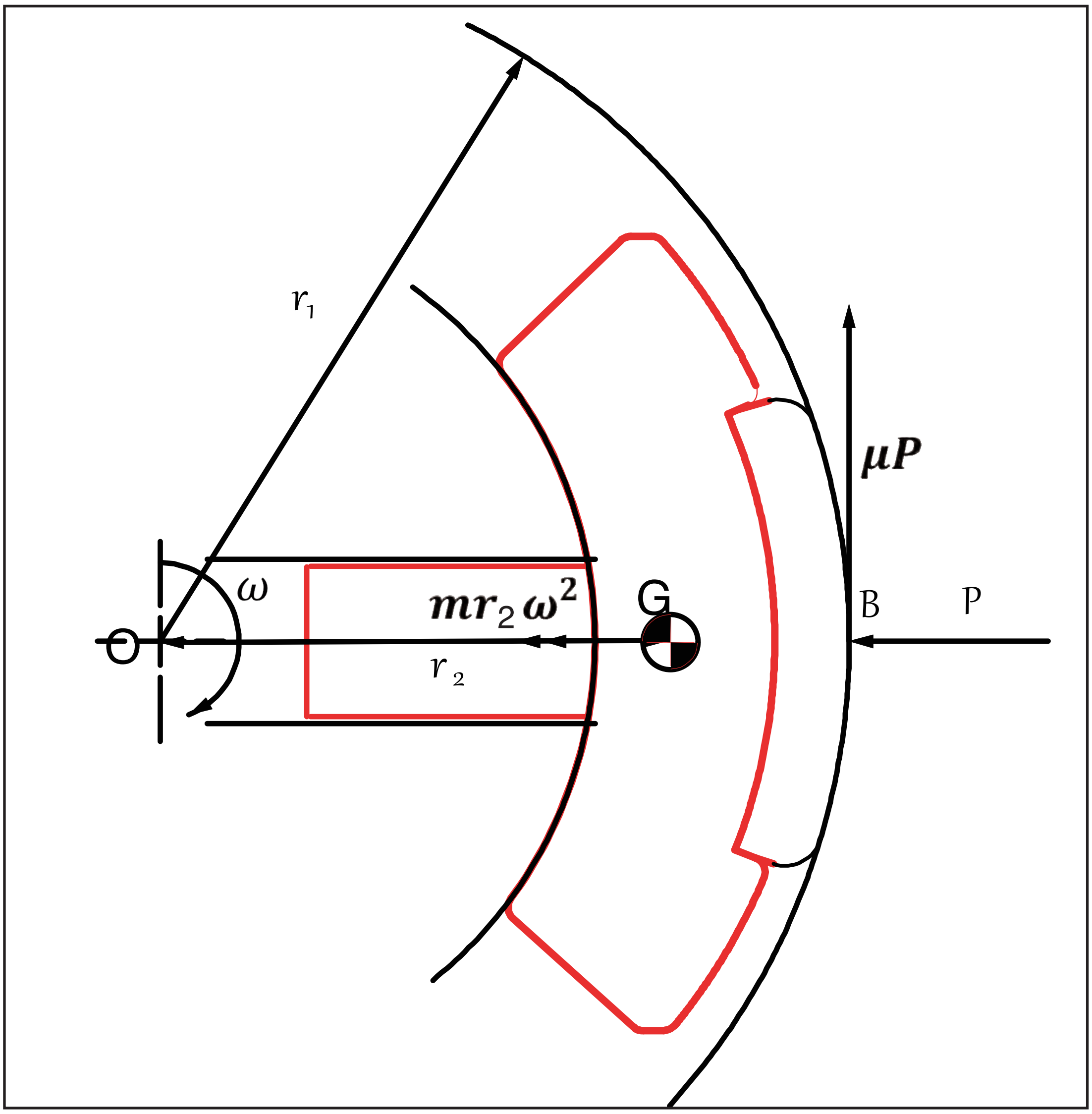


figure 7.4 Force diagram - sliding brake simplified

Taking the detail from figure 7.4 and the information from part one of this analysis.

Resolving radially, equating the effective force to the applied force we have:-

$$P = mr_1\omega^2 \text{.................................}(7.9)$$

The brake torque = $\mu P r_1 = m r_1 r_2 \mu \omega$

If there are four shoes and the angular shoe velocity is N rev/s, then:

$$\text{since} \quad \omega = 2\pi N \text{............................}(7.10)$$

and, $\therefore$ Brake torque $= 16\pi^2 m r_1 r_2 \mu N^2$......(7.8)

Based on figure 7.4 and the shoe data given in part one of the analysis, the brake torque for the sliding brake design is:

$$\text{Brake torque} = 12.3 \times 10^{-3} \times N^2 \quad \text{(Nm)}$$

7.3 Analysis of the two pivoting brake shoe design

The design comprises 2 brake shoes that can be configured as 2 leading (L) or 2 trailing (T) or one leading and one trailing (L+T). It is assumed that the resultant radial brake pad force acts at the brake pad centre.

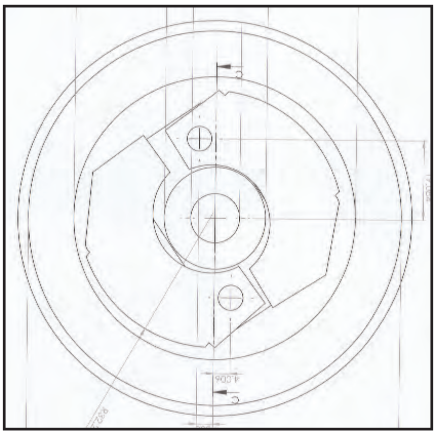


figure 7.5 :Two shoe schematic

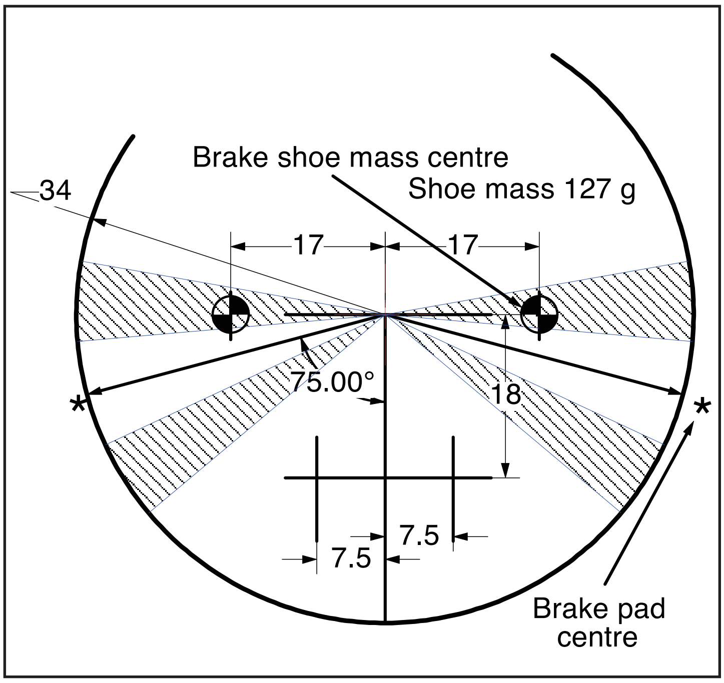


figure 7.6 : Main brake shoe dimensions

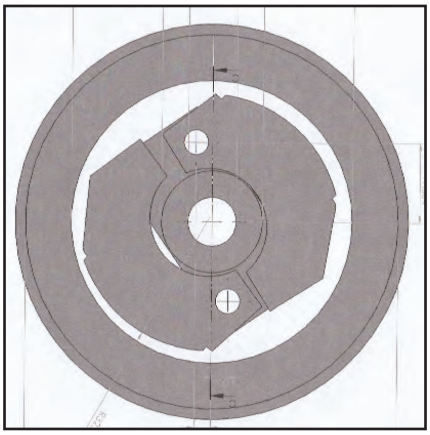


figure 7.7 : Two shoe negative

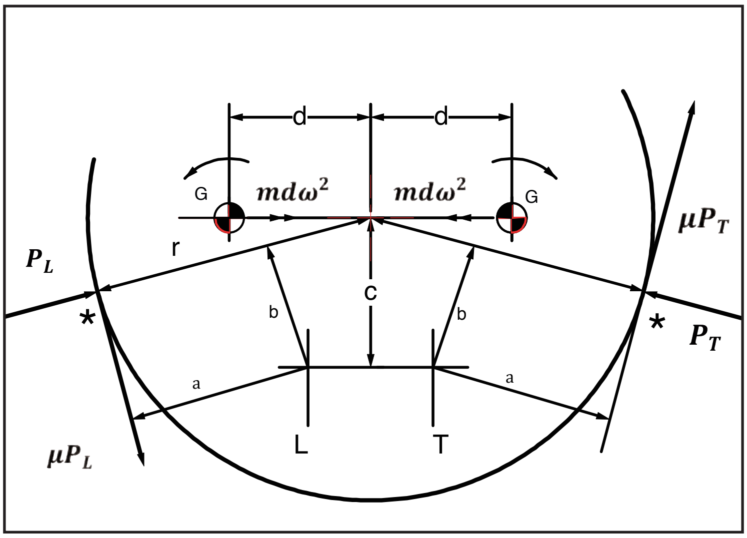


figure 7.8 : Force vector diagram

Equating effective mass acceleration moments to applied moments we have:

Moments about L $\quad md\omega^2 \times c = P_L \times b - \mu P_L \times a \;\;\therefore\;\; P_L = \dfrac{md\omega^2 c}{b - \mu a}$

Moments about T $\quad md\omega^2 \times c = P_T \times b + \mu P_T \times a \;\;\therefore\;\; P_T = \dfrac{md\omega^2 c}{b + \mu a}$

Equation (7.2.2):

Braking torque for $L + T = \mu(P_L + P_T) \times r = \mu m d \omega^2 c r\left(\frac{1}{b-\mu a} + \frac{1}{b+\mu a}\right)$

$$2\pi N = \omega \ \ and \qquad \left(\frac{1}{b-\mu a} + \frac{1}{b+\mu a}\right) = \frac{2b}{b^2 - \mu^2 a^2} \qquad (7.3.3)$$

$$\therefore \text{ Brake torque for } L + T = \frac{8\pi^2 mdcrb\mu}{b^2 - \mu^2 a^2} \times N^2 \qquad (7.3.4)$$

$$\text{Brake torque for 2L} = \frac{8\pi^2 mdcr\mu}{b - \mu a} \times N^2 \qquad (7.3.5)$$

$$\text{Brake torque for 2T} = \frac{8\pi^2 mdcr\mu}{b + \mu a} \times N^2 \qquad (7.3.6)$$

Considering the information in *figure 7.5,7.6,7.7* and *figure 7.8* where:

$$a = 0.023 \text{ m}; b = 0.016 \text{ m}; c = 0.018 \text{ m}; d = 0.017 \text{ m}; r = 0.034 \text{ m}$$
$$m = 0.127 \text{ kg}: \mu = 0.35$$

Where L = leading and T = Trailing

$$\text{Brake torque for 2L} = \frac{0.0104\mu N^2}{1.6-2.3\mu} \text{ Nm} = 4.58 \times 10^{-3} N^2 \text{ Nm}$$

$$\text{Brake torque for L+T} = \frac{0.0167\mu N^2}{2.56-5.29\mu^2} \text{ Nm} = 3.06 \times 10^{-3} N^2 \text{ Nm}$$

and,

$$\text{Brake torque for 2T} = \frac{0.0104\mu N^2}{1.6+2.3\mu} \text{ Nm} = 1.51 \times 10^{-3} N^2 \text{ Nm}$$

7.4 Analysis of 4 pivoting brake shoe design

The configuration comprises 4 brake shoes that can pivot in order that the shoes engage with the brake band. The arrangement of the shoes can be 4 leading, 4 trailing or 2 leading and 2 trailing. A direct comparison can be made between the pivoting shoes and the sliding shoes, figures 7.9,7.10,7.11 and 7.12.

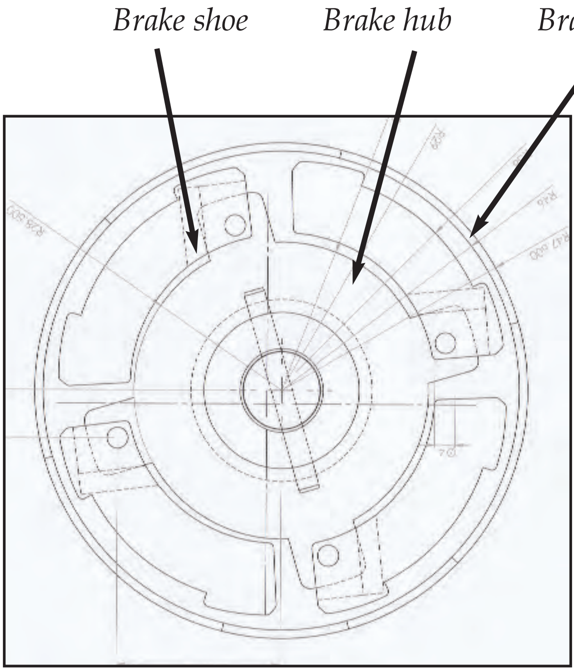


figure 7.9: 4 shoe arrangement which is set to either trailing or leading in outline schematic

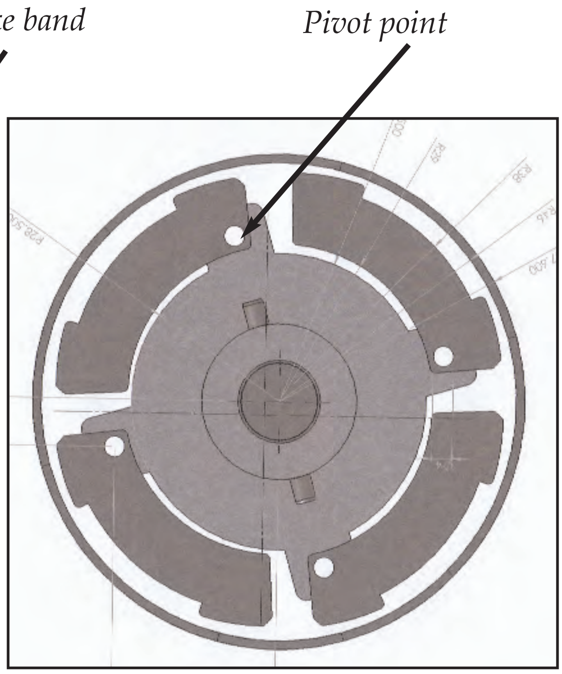


figure 7.10 : 4 shoe arrangement which is set to either trailing or leading shadow format

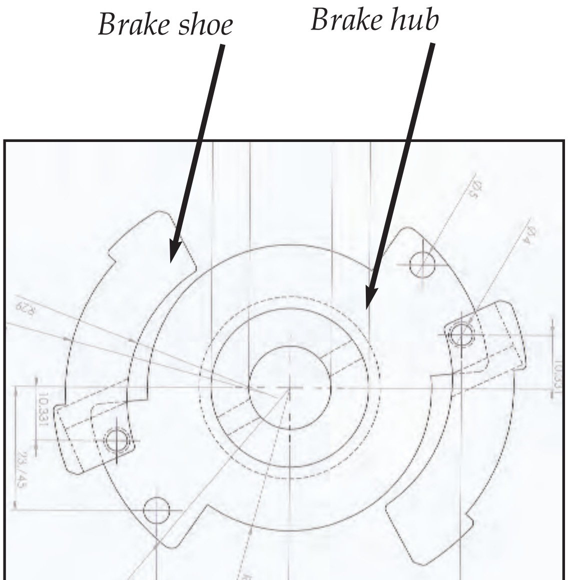


figure 7.11 : 4 shoe arrangement which is set with 2 leading and 2 trailing outline schematic

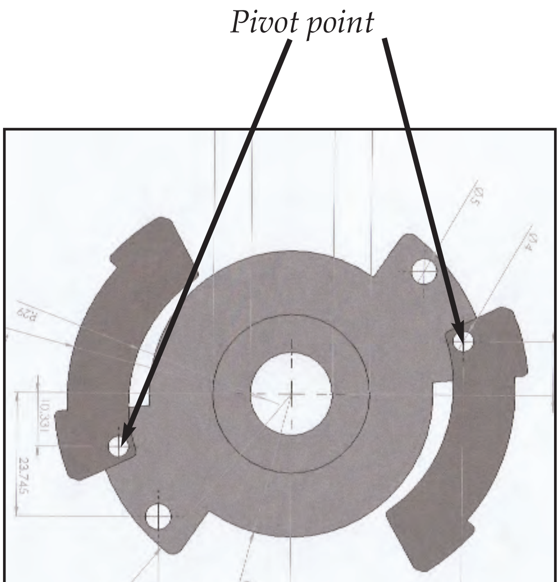


figure 7.12 : 4 shoe arrangement which is set with 2 leading and 2 trailing shadow format

Figure 7.13 shows a drawing of a pivoting brake shoe arrangement from the concept design to a scale of 2:1. It is shown with two alternative pivot points at A_1 and A_2. The centre of mass of the brake shoe is shown as point G and the centre of brake band at 0.

The brake band has a radius r, B marks the centre of the brake pad arc and the radial forceP is the resultant force on the shoe between the brake band and the brake pad and it is assumed to pass through B.

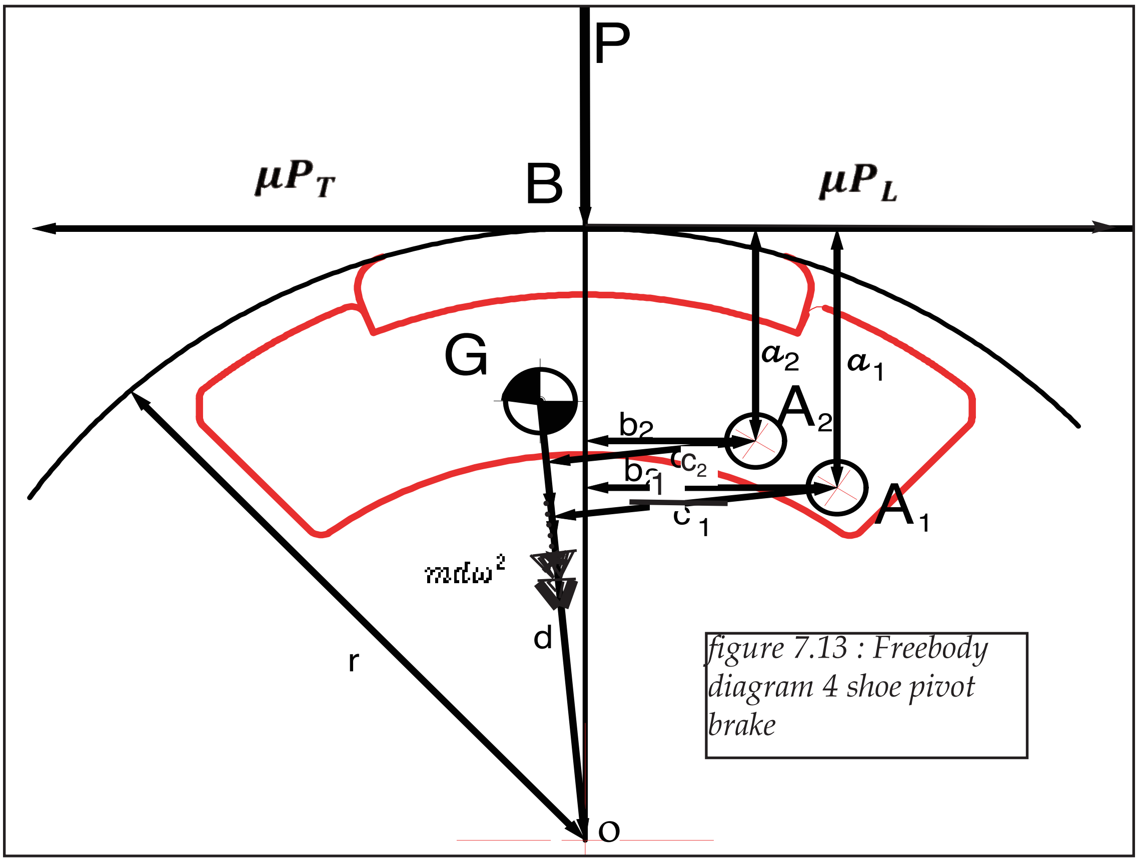


figure 7.13 : Freebody diagram 4 shoe pivot brake

Tangential forces

μP_L to the right for a leading shoe

μP_T to the left for a trailing shoe

The forces represent the frictional force on the shoe, L representing a leading shoe with the shoe rotating anticlockwise and T representing a trailing shoe rotating clockwise.

The radial mass acceleration effective force at G is:

$md\omega^2$

Where, m is the mass of the shoe

d is the radial distance from G to 0 and,

ω is the angular velocity of the shoe in radians per second

Taking moments about $A_{1,2}$, equating effective moments to applied moments we have:-

$$md\omega^2 \times c = Pb \pm \mu Pa \qquad (7.4.2)$$

Where – is a leading shoe and + is a trailing shoe.

The friction torque at the pivot pin is negligible.

Therefore, the brake torque: $\mu Pr = \frac{mdcr\mu\omega^2}{b \pm \mu a}$ (7.4.3)

If the angular velocity of the brake shoe is:

$N \text{ rev/s}, 2\pi N = \omega$

Therefore, the brake torque: $= \frac{4\pi^2 mdcr\mu N^2}{b \pm \mu a}$ (7.4.4)

The denominator of the equation for the leading shoe is:

$$(b - \mu a) \qquad (7.4.5)$$

This can be zero if

$$\mu = \frac{b}{a} \qquad (7.4.6)$$

However, this would result in an infinite torque which is a condition referred to as spragging and must be avoided.

Spragging occurs when the resultant of:

$$P \text{ and } \mu P_L \qquad (7.4.7)$$

passes through the pivot point A

The brake torques for the four brake shoes are:

Brake torque for 4T $= \frac{16\pi^2 mdcr\mu}{b + \mu a} \times N^2$ (7.4.8)

Brake torque for 4L $= \frac{16\pi^2 mdcr\mu}{b - \mu a} \times N^2$ (7.4.9)

For the 2L + 2T configuration then:

Brake torque for 2L + 2T = $\frac{16\pi^2 mdcrb\mu}{b^2 - \mu^2 a^2} \times N^2$ (7.4.10)

Note: $\frac{1}{b - \mu a} + \frac{1}{b + \mu a} = \frac{2b}{b^2 - \mu^2 a^2}$ (7.4.11)

From figures 7.9 – 7.12 the brake shoe data is:

$r = 0.046$ m; $d = 0.033$ m;

$a_1 = 0.0195$ m; $b_1 = 0.0175$ m; $c_1 = 0.0195$ m

$a_2 = 0.016$ m; $b_2 = 0.011$ m; $c_2 = 0.0135$ m

m=0.085 kg; μ=0.4

Therefore, the brake torques are:

4 leading shoes pivot point $A_1 = 16.38\times10^{-3}\times N^2$ Nm

2 leading+2 trailing shoes pivot point $A_1 = 11.33\times10^{-3}\times N^2$ Nm

4 trailing shoes pivot point $A_1 = 6.28\times10^{-3}\times N^2$ Nm

4 leading shoes pivot point $A_2 = 23.92\times10^{-3}\times N^2$ Nm

2 leading+2trailing shoes pivot point $A_2 = 15.12\times10^{-3}\times N^2$ Nm

4 trailing shoes pivot point $\mathrm{A}_2 = 6.32\times10^{-3}\times\mathrm{N}^2$ Nm

From this data the spragging coefficient of friction for point A_1 is about 0.9 and for pivot point A_2 is about 0.7, both of which are considerably above the actual coefficient of friction being 0.4(obtained from supplier specification sheet), so no spragging will occur

Figures 7.14 and 7.15 shows the theoretical results for both sliding and pivoting shoes.

For the 4 leading shoe brake it can be seen that the pivot point A_2 produces about 46% higher torque than pivot point A_1 whereas for the 4 trailing shoes the torques are nearly the same.

Comparing these pivoting brake shoes with the sliding shoe brake it can be seen that the 2 Leading + 2 Trailing shoe brakes using pivot point A_1 produces a similar torque to the sliding brake even though their brake shoe mass is considerably less (32%). Again from Fig 93 it can be seen that the results for the power produced by the pivot point A_2 is more than double that produced by the sliding shoes. Also the power produced for the 2 Leading + 2 Trailing shoes brakes pivoting at A_1 are similar to the sliding brake.

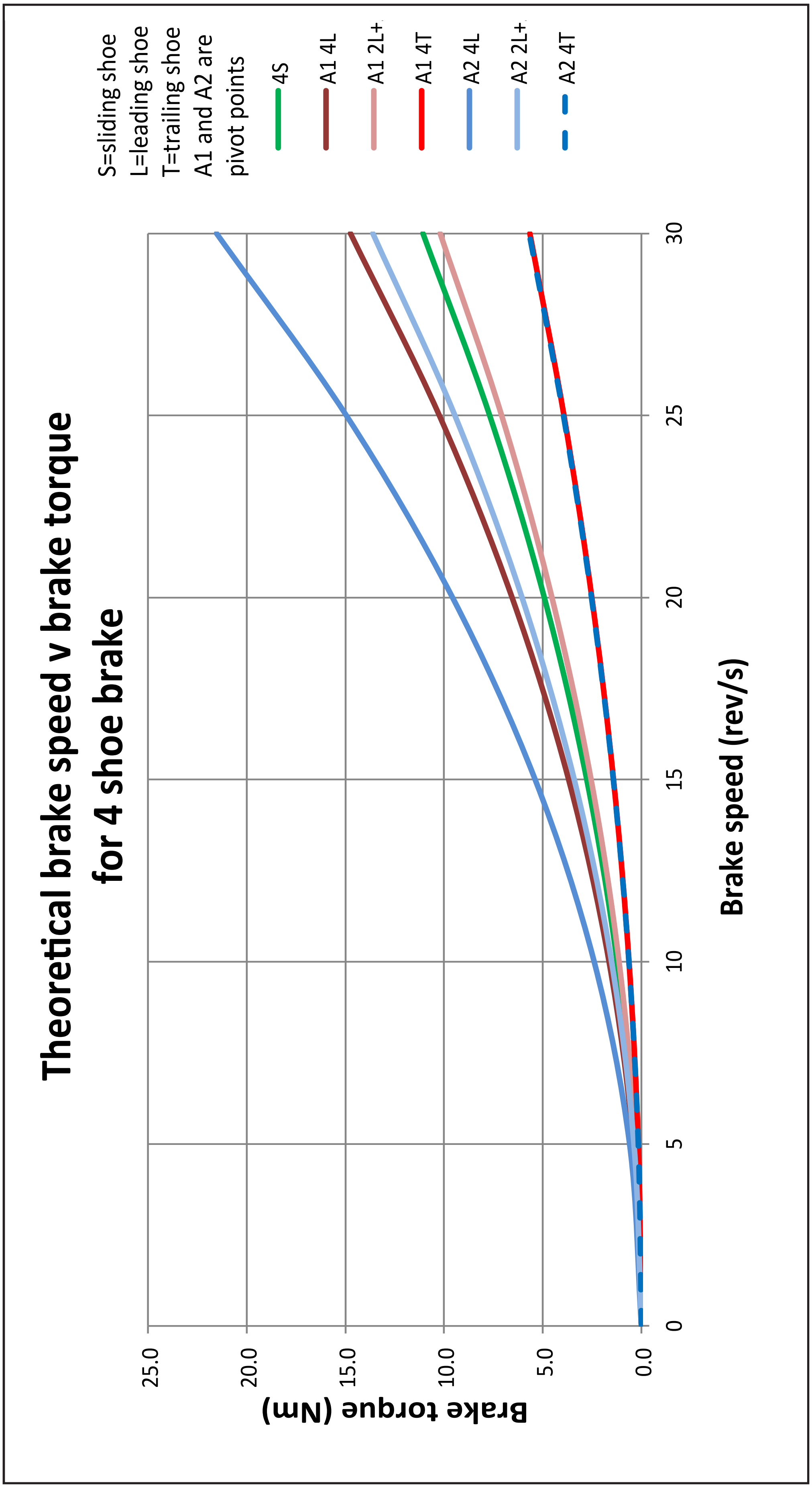


figure 7.14: Theoretical brake speed against brake torque for 4 pivoting shoes and 4 sliding shoes

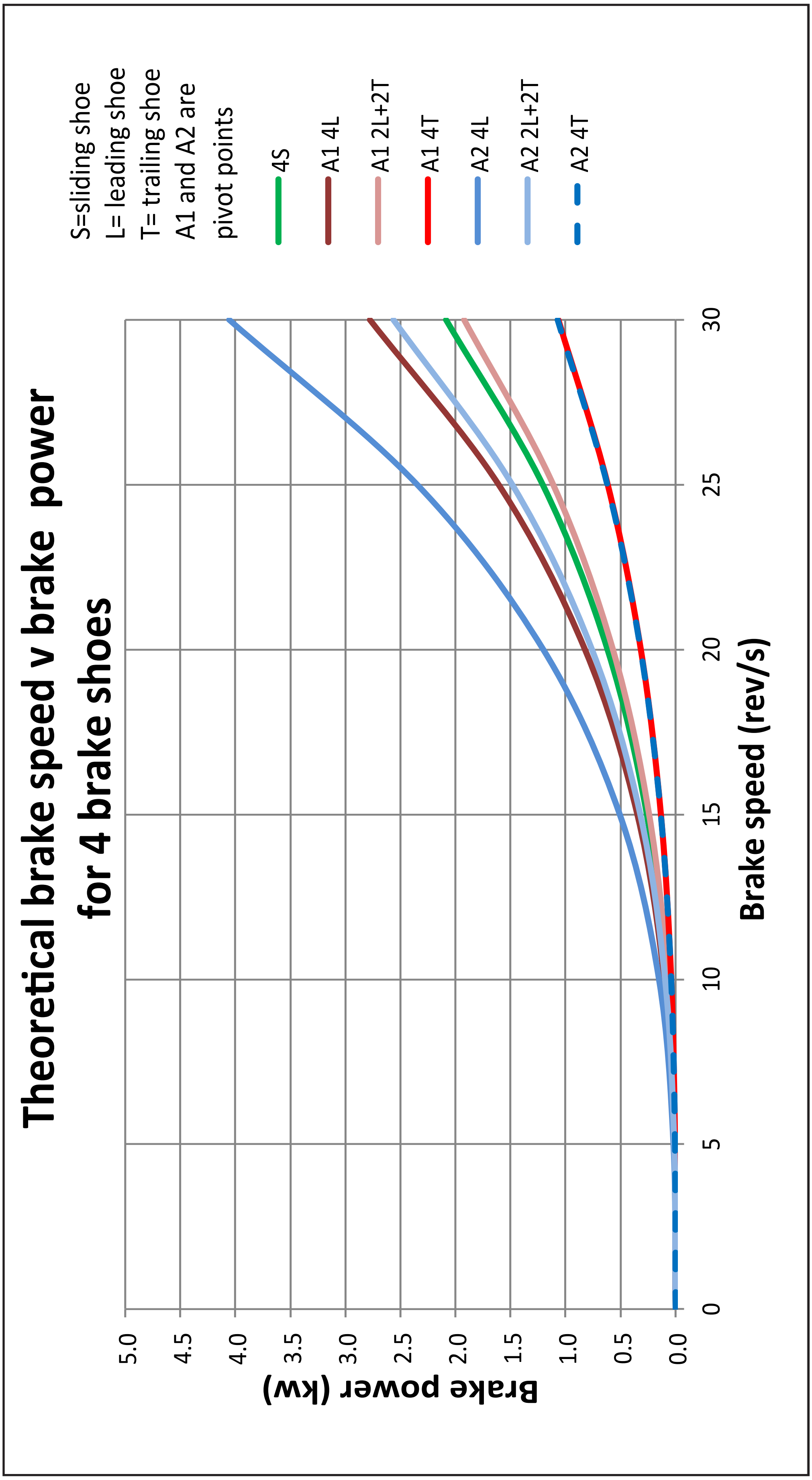


figure 7.15: Theoretical brake speed against brake power for 4 pivoting shoes and 4 sliding shoes

7.5 Analysis of the tape reel descender

A factor that can be used to influence the descent speed is the variable radius on the spool as the tape pays out from the start radius to the finish radius.

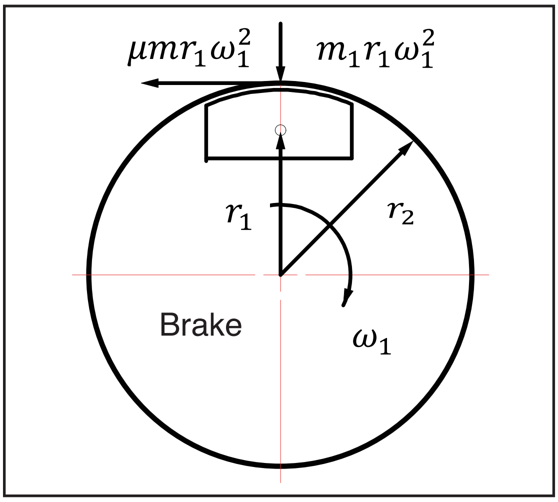


figure 7.16: Brake shoe force diagram

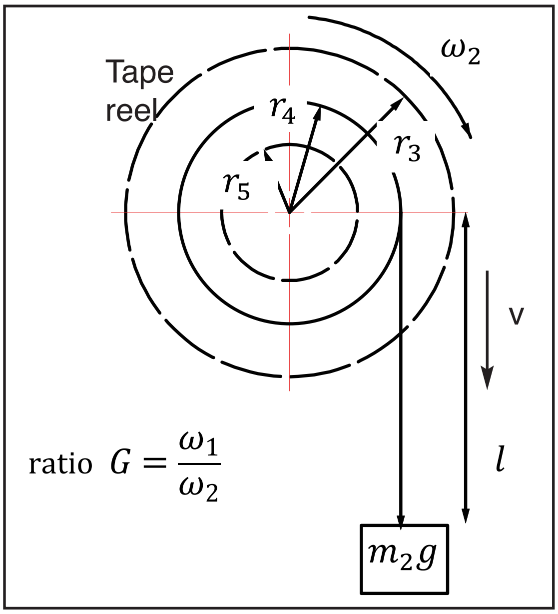


figure 7.17 : Tape reel gearing diagram

This theory neglects the very low deceleration of the mass, which reduces in velocity from 2.5 m/s to zero in 60 seconds or more (see figures 7.18 and 7.19). This equates to an average deceleration of about 0.04 m/s^2 or less.

Radial force on one sliding brake shoe = $mr_1\omega_1^2$ (7.5.1)

Torque due to 4 sliding brake shoes = $4\mu m_1 r_1 r_2 \omega_1^2$ (7.5.2)

r_3 = tape start radius

r_5 = tape end radius

r_4 = tape radius at drop length l

Output torque from geared down spool shaft = $4\mu m_1 r_1 r_2 \omega_1^2 \times \frac{G}{\eta} = m_2 g r_4$ ..(7.5.3)

Where η = gear efficiency

By equating the area of the spool from r_3 to r_4 to the edge area of tape of length l then:

$\pi(r_3^2 - r_4^2) = lt$ where t = tape thickness……………………………………(7.5.4)

$$v = r_4\omega_2 = r_4 \frac{\omega_1}{G} \quad \therefore \omega_1 = \frac{vG}{r_4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.5)$$

Substituting ω_1 from equation 3 into equation 1

$$\therefore 4\mu m_1 r_1 r_2 \times \frac{v^2}{r_4^2} G^2 \times \frac{G}{\eta} = m_2 g r_4 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.6)$$

From equation 2: $r_4 = \left(r_3^2 - \frac{lt}{\pi}\right)^{\frac{1}{2}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.7)$

From equation 4; $v^2 = \frac{m_2 g r_4^3 \eta}{4\mu m_1 r_1 r_2 G^3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.8)$

substituting for r_4^3 in equation 7.5.8 from equation 7.5.7 then:

Hence $v^2 = \frac{m_2 g \eta}{4\mu m_1 r_1 r_2 G^3} \times \left(r_3^2 - \frac{lt}{\pi}\right)^{\frac{3}{2}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.9)$

When $l = 0$ *then* $v_o^2 = \frac{m_2 g \eta r_3^3}{4\mu m_1 r_1 r_2 G^3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.10)$

When $l = l_{max}$ *then* $v_1^2 = \frac{m_2 g \eta r_5^3}{4\mu m_1 r_1 r_2 G^3} \ldots\ldots\ldots\ldots\ldots\ldots (7.5.11)$

Where $l_{max} = \frac{\pi}{t}\left(r_3^2 - r_5^2\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7.5.12)$

Equation 7.5.9 has to be modified to account for the friction of the tape passing over the entry guide rod (see sections 8.2 and 8.3)

$$v^2 = \frac{g\eta}{4\mu m_1 r_1 r_2 G^3} \times \left(r_3^2 - \frac{lt}{\pi}\right)^{\frac{3}{2}} \times m_2 \left[a + (b - a)\frac{l}{l_{max}}\right] \ldots\ldots\ldots\ldots\ldots (7.5.13)$$

From equation 8 the drop velocity can be calculated for any drop length l and drop mass m_2

Where $l_{max} = \frac{\pi}{t}\left(r_3^2 - r_5^2\right)$

$\mu = 0.4$; $r_1 = 0.034$ m; $r_2 = 0.046$ m; $r_3 = 0.105$ m; $r_5 = 0.025$ m; $G = 8$

$m_1 = 0.125$ kg; $\eta = 0.68$*; $t = 0.5\times10^{-3}$ m

* For η *refer to section* 8.1

$$\therefore l_{max} = \frac{\pi}{0.5\times10^{-3}}\left(0.105^2 - 0.025^2\right) = 65.3 \text{ m}$$

From the data derived in the test to determine the coefficient of friction between the tape and the guide rod chapter 8.3 $a = 0.85$ and $b = 0.97$

$$a = 0.85 \text{ and } b = 0.97$$

$$\therefore v^2 = \frac{9.81\times0.68}{4\times0.4\times0.125\times0.034\times0.046\times8^3}\times\left(0.105^3 - \frac{l\times0.5}{\pi\times10^3}\right)^{\frac{3}{2}}\times m\left(0.85 + \frac{0.12l}{65.3}\right)$$

$$v = \left[41.6\times\left(0.105^3 - \frac{0.157l}{1000}\right)^{\frac{3}{2}}\times m_2\left(0.85 + \frac{1.83l}{1000}\right)\right]^{\frac{1}{2}}$$

The graph of drop length against drop velocity is shown in figure 7.18 for 3 masses

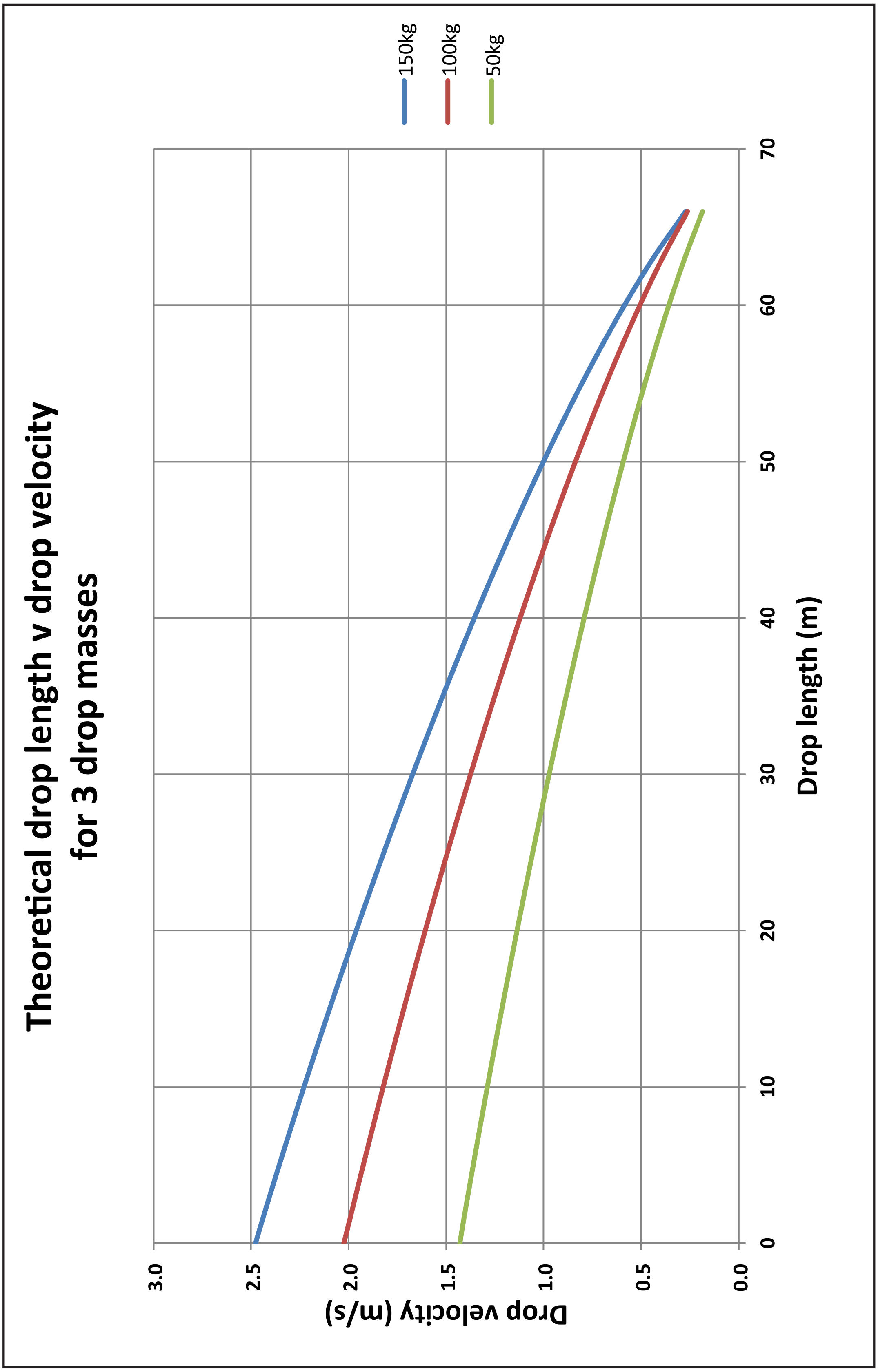


figure 7.18 : Drop length against drop velocity

A plot of the drop time against the drop distance can be obtained using the drop distance against the drop velocity data.

$$v = \frac{dl}{dt} \quad \therefore \int_{t_1}^{t_2} dt = \int_{l_1}^{l_2} \frac{dl}{v}$$

$$\therefore \; t_2 - t_1 = \left(\frac{1}{v}\right)_{average} (l_2 - l_1)$$

$$t_2 - t_1 = \frac{1}{2}\left(\frac{1}{v_1} + \frac{1}{v_2}\right)(l_2 - l_1)$$

This is the time taken to travel the distance $l_1 \; to \; l_2$

To obtain the cumulative time to go from $l = 0 \; to \; l = l_2$ the individual time increments are added together. This can easily be achieved using Excel.

The graph of drop time against drop length is shown in figure 97.

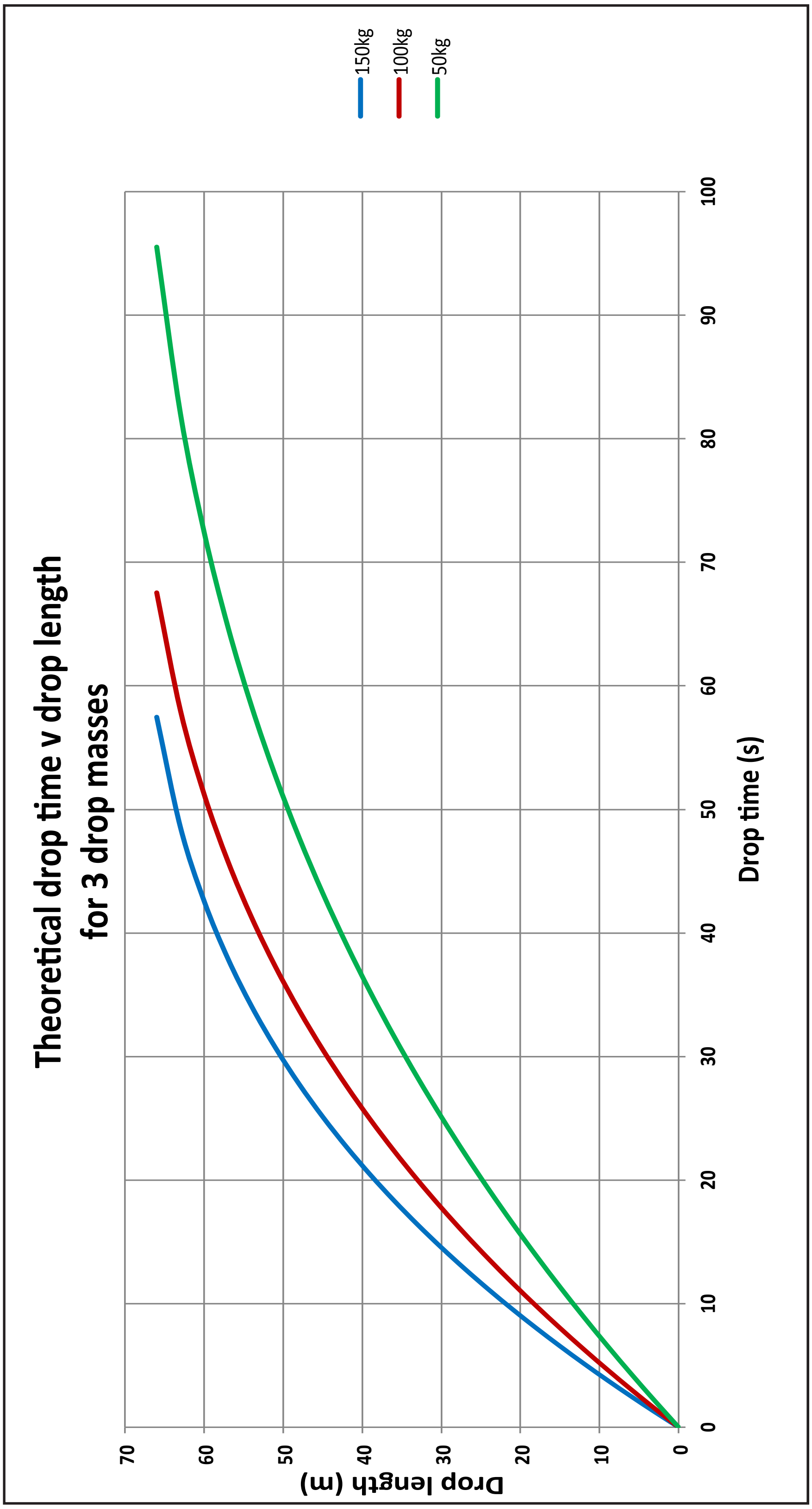


figure 7.19 : Drop time against length

7.6 Discussion and summary

The formulae given in this chapter give an analysis for each of the designs of brake that are considered. The diagrams show the forces and moments on each brake shoe in order to determine the brake torque developed.
The brake shoe equations are then used to produce graphs of brake speed against torque for all the brake shoe arrangements.
The final equations enable graphs to be plotted of drop distance against drop velocity and drop time against drop distance for a range of drop masses.
These curves show that it is possible to control both the initial and final drop velocities by careful selection of the maximum and minimum spool radii.
In the next section these theoretical results are compared with results obtained using the brake test rig and the drop test rig.

Chapter 8 : Testing the designs

8.1 Introduction

In the following chapter the designs are evaluated by a number of tests and comparison is drawn with theoretical analysis. In order to evaluate the designs several test rigs were developed and manufactured for efficiency, brake performance, static strength and friction. The National lift tower, Northampton, SATRA technology, Liverpool bulk coal terminal and AYD test towers were used together with the instrumentation to test various aspects of the design. The chapter considers the set up, instrumentation and results obtained during the program of work aimed at proving the designs..

8.2 Test to determine the tranmission efficiency of the brake

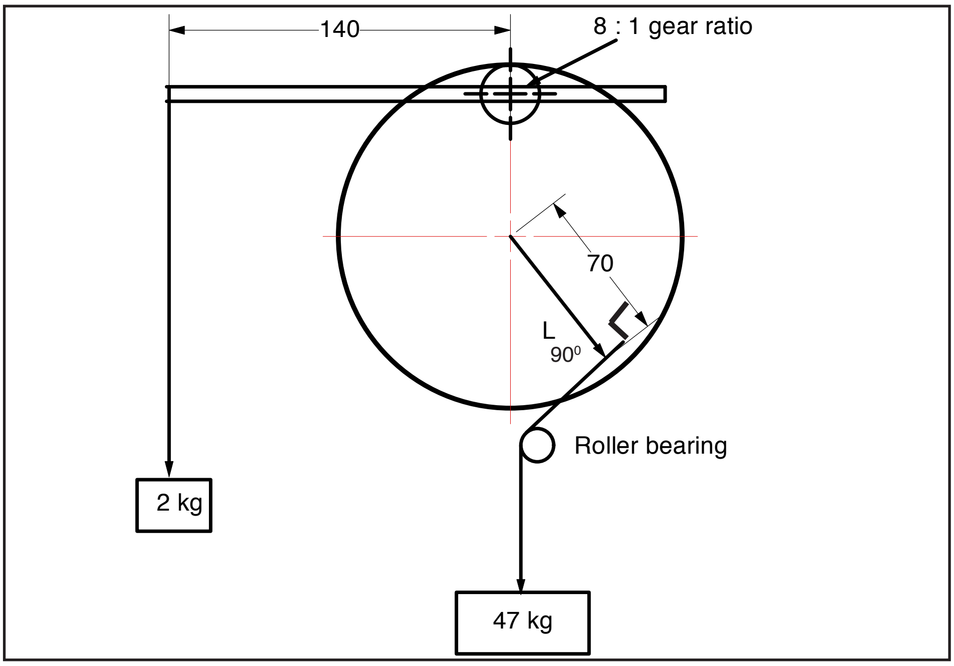


figure 8.1: Transmission efficiency diagram

Figure 8.1 shows a schematic drawing of the test set up, the test itself is shown in the figure 8.2.

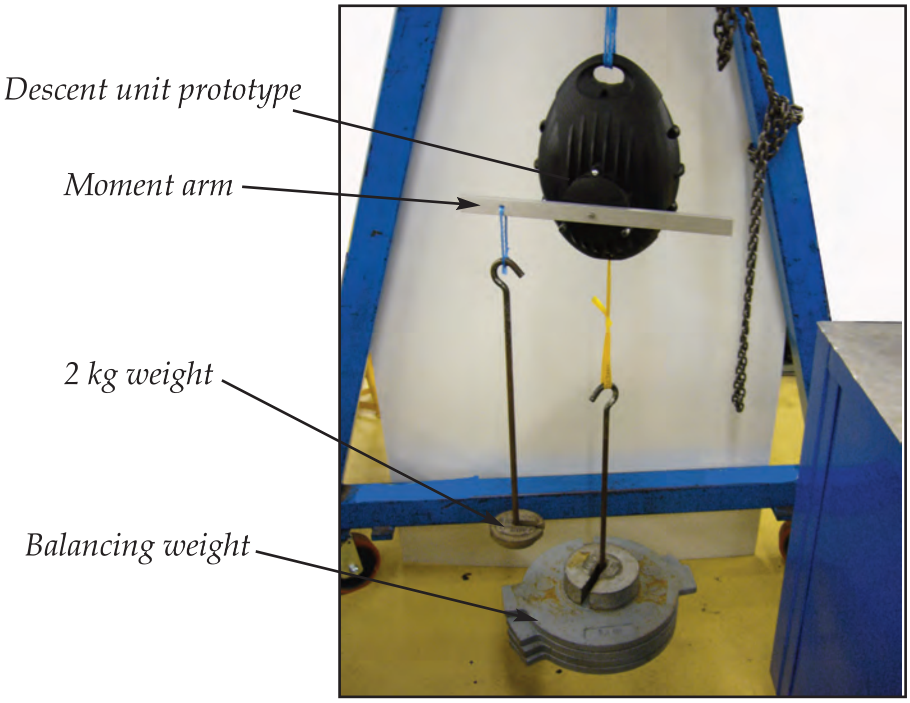


figure 8.2 : Transmission efficiency set up

A 2 kg weight was suspended on a 140 mm moment arm connected to the brake pinion shaft, on the end of the shaft was a 9 tooth pinion which is engaged with a 72 tooth ring gear mounted on the drum inside the unit. A tape was attached at a radius of 70 mm and passed over a roller bearing. A weight was attached to the free end of the tape.

For perfect balance the weight to be applied to the free end of the tape is determined by the following formulae:

2 kg x 2 (moment ratio) x 8 (gear ratio) = 32 kg

This would achieve perfect balance if the efficiency was 100%. But it was found that a weight of 47 kg was required in order to cause the 2 kg weight to rise up.

The transmission efficiency was therefore $\frac{32}{47}$ =0.68 or 68%

The inefficiency found could in part be put down to friction in the gear chain and also sticking within the device as the load is applied.

8.3 Modification to account for the friction of the tape passing over the entry guide rod

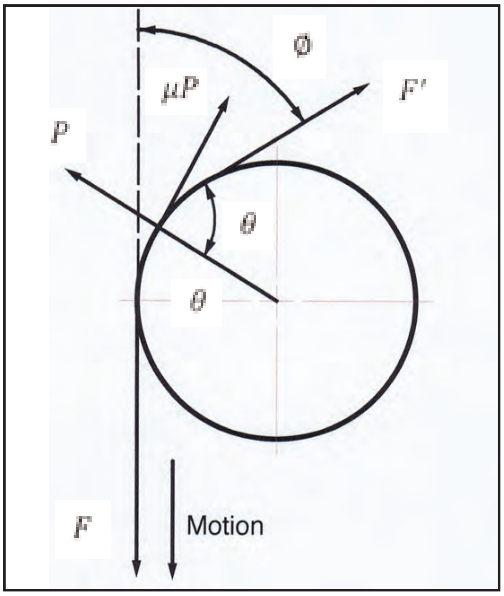


figure 8.3: Modification diagram for friction

From figure 8.3

$$\emptyset + 2\theta = 180^0 \;\therefore\; \theta = 90 - \frac{\emptyset}{2} \tag{8.3.1}$$

$$P = (F + F')cos\theta \;\; and \;\; F = \mu P + F' \tag{8.3.2}$$

$$\therefore \; F' = F - \mu P = F - \mu F cos\theta - \mu F' cos\theta \tag{8.3.3}$$

$$F'(1 + \mu cos\theta) = F(1 - \mu cos\theta) \tag{8.3.4}$$

$$F' = \frac{F(1 - \mu cos\theta)}{(1 + \mu cos\theta)} = F \times k \tag{8.3.5}$$

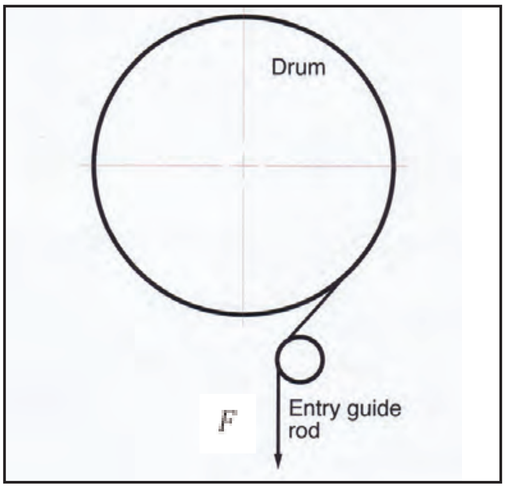


figure 8.4 : Modification for friction over entry guide rod

At $l = 0$ let $k = a$ At $l = l_{max}$ let $k = b$

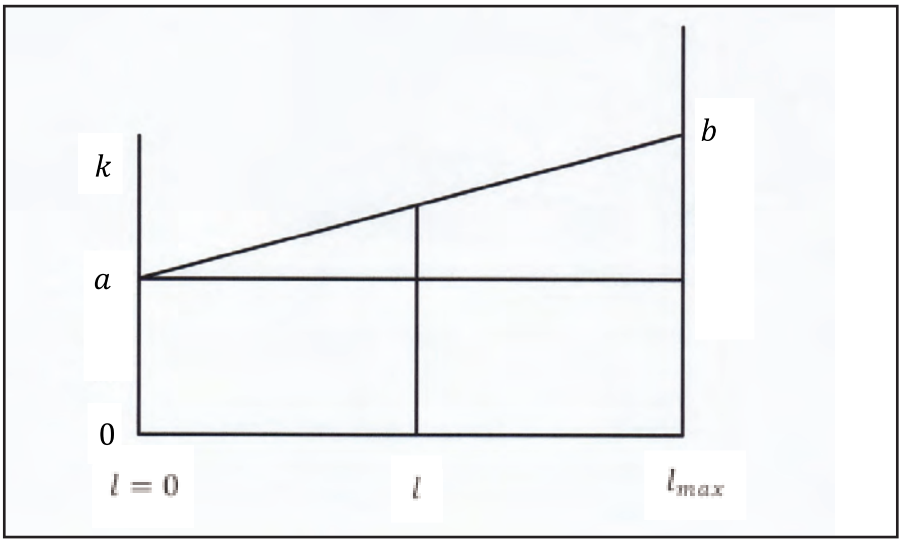


figure 8.5: Resultants bar diagram

Therefore from figure 8.5

At $l = 0$ let $k = a$ At $l = l_{max}$ let $k = b$ (8.3.6)

$At\ l \quad k = \left[a + (b - a)\frac{l}{l_{max}}\right]$ (8.3.7)

From figure 8.6 we have

Assuming a linear relationship between $k$and l

From figure 8.5:

$\emptyset_{l=0} = 48^0 \;\therefore\; \theta_{l=0} = 66^0 \quad cos66^0 = 0.41$

$\phi_{l=lmax} = 8^0 \;\therefore\; \theta_{l=lmax} = 86^0 \quad cos86^0 = 0.07$

$\mu = 0.2 \;\therefore a = \dfrac{1 - 0.2\times0.41}{1 + 0.2\times0.41} = 0.85$

and $b = \dfrac{1 - 0.2\times0.07}{1 + 0.2\times0.07} = 0.97$

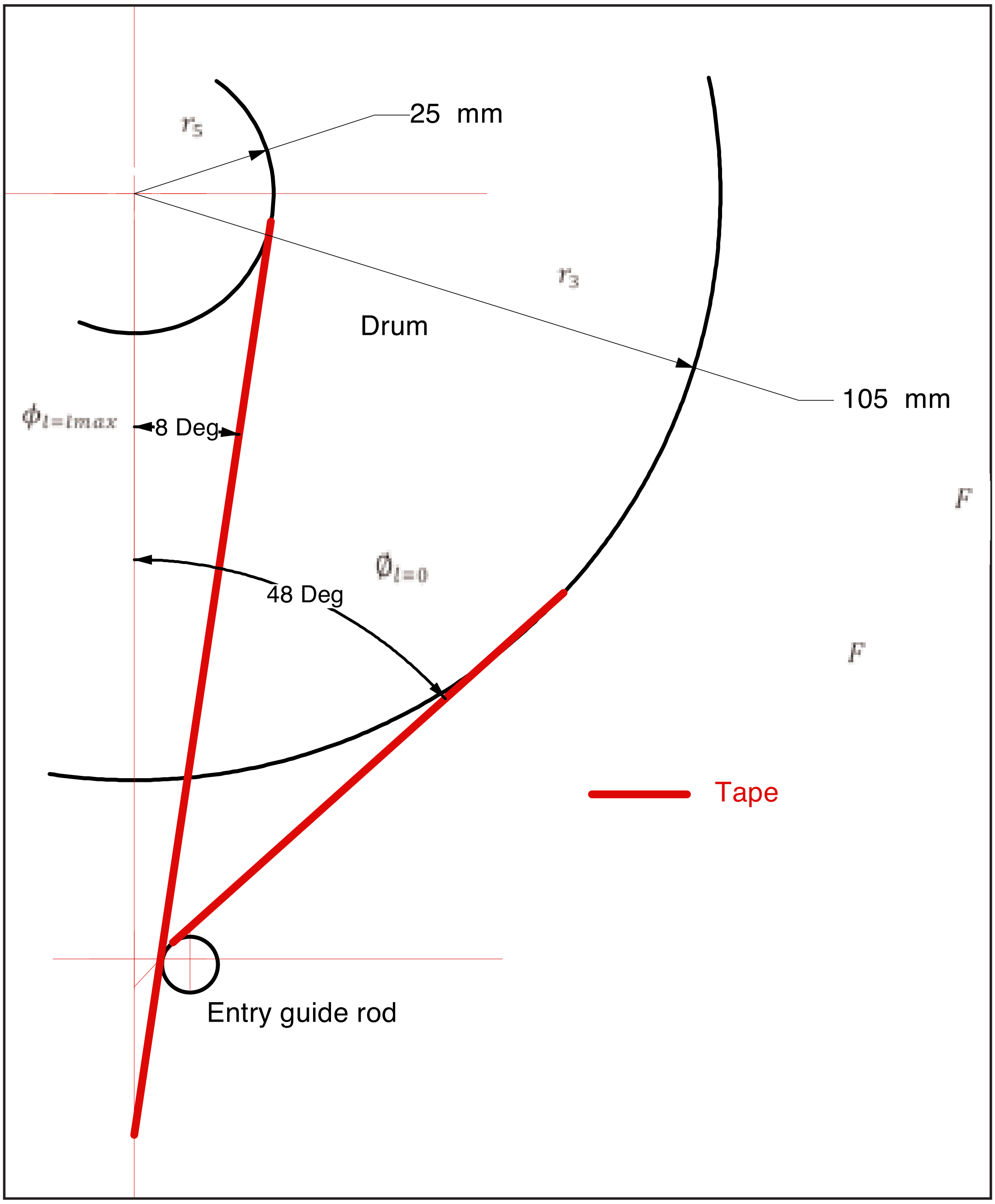


figure 8.6: Force diagram to scale for layout

8.4 Test to determine the coefficient of friction between the tape and the guide rod

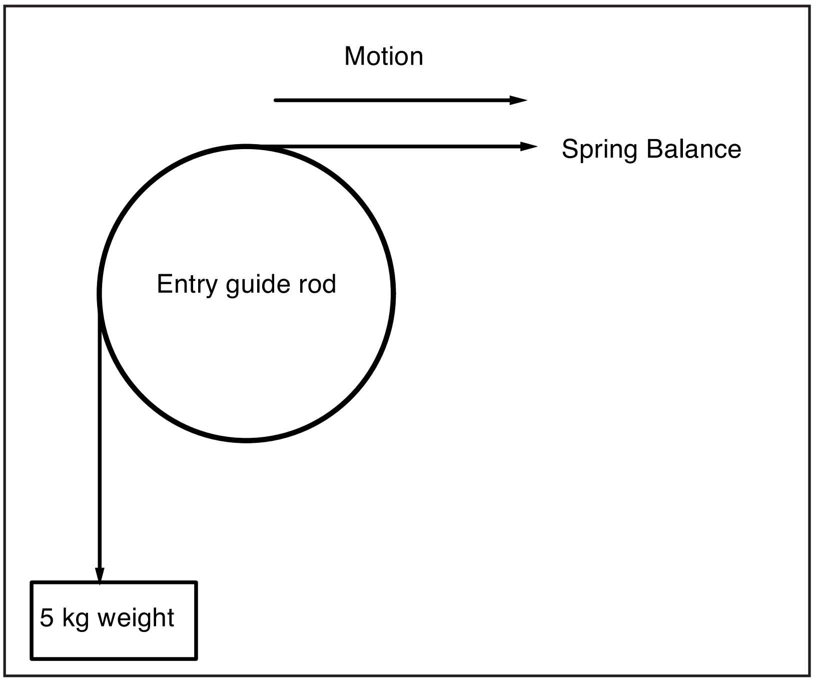


figure 8.7 : Coefficient of friction between tape and guide rod diagram

A spring balance was used to apply a horizontal force to the tape which wrapped around 90 Deg. of the entry guide rod. A mass of 5 kg was suspended on the other end of the tape as shown in figure 100.

The force required to lift the weight was 6.7 kg as indicated on the spring balance.

Using the formula from section 8.3:

$$F' = F\frac{(1 - \mu cos\theta)}{(1 + \mu cos\theta)} \quad \text{where } \theta = 45^0$$

$F' = 5$ kg and $F = 6.7$ kg

$$\therefore 5 = 6.7 \times \frac{(1 - 0.7\mu)}{(1 + 0.7\mu)}$$

$$\therefore 5 + (5 \times 0.7\mu) = 6.7 - (6.7 \times 0.7\mu)$$

$$\therefore 1.7 = 11.7 \times 0.7\mu \quad \therefore \mu = 0.2$$

8.5 Brake torque test rig.

The brake test rig was designed to measure the rotational speed and torque produced by the centrifugal brake units. The complete test rig is shown in figure 8.8.

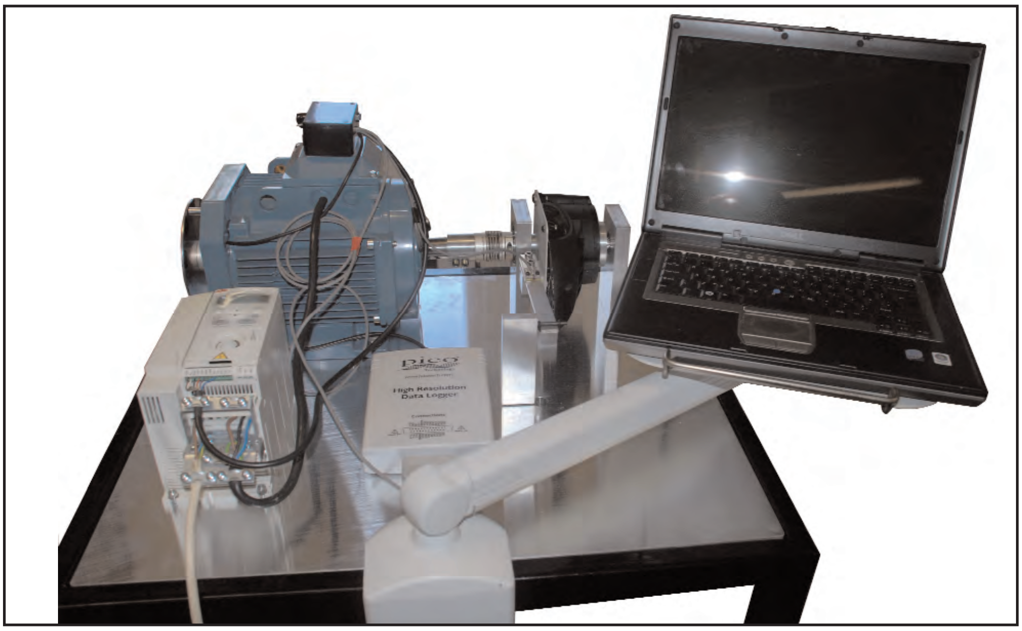


figure 8.8 : Brake torque test rig

The rotational speed is measured by a reflective optical switch which is looked at a disc mounted on the shaft of the motor driving the brake(the motor fan was removed). The disc surface was polished and then painted with matt black stripes as shown in the photograph figures 8.9 and 8.10.

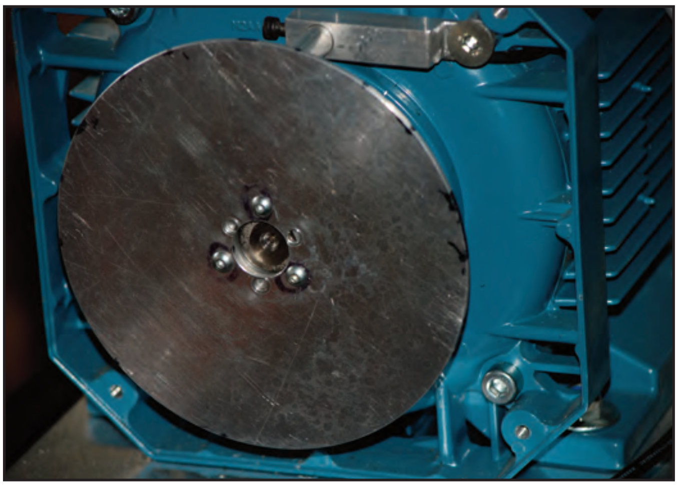


figure 8.9 : Rotational speed disc on motor

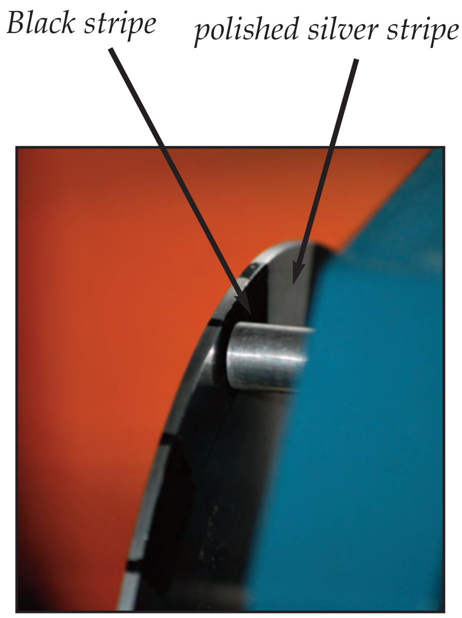


figure 8.10 : Opto switch shown reading black marks

The output from the optical switch was fed into a comparator which gave +/- 5 v square wave output when the switch passed from a polished to a matt black segment on the surface. This square wave was fed into a frequency to voltage converter which gave a voltage output proportional to the rotational speed of the motor shaft. Figure 108 shows a block diagram of the circuitry.

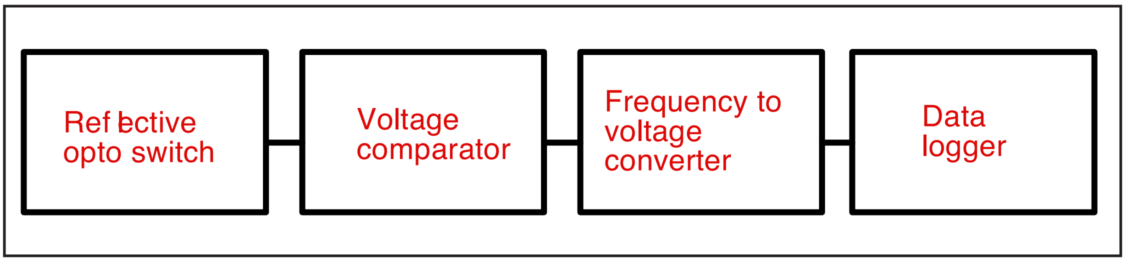


figure 8.11 Bock diagram for circuitry

A trimming resistor used to adjust the output voltage to be less than 2.5 volts for the highest motor speed. 2.5 volts is the maximum allowable input voltage to the Pico high resolution data logger.
The brake test rig was designed so that the brake case is supported on bearings and prevented from rotating by a torque arm that has strain gauges on as shown in figure 8.12.

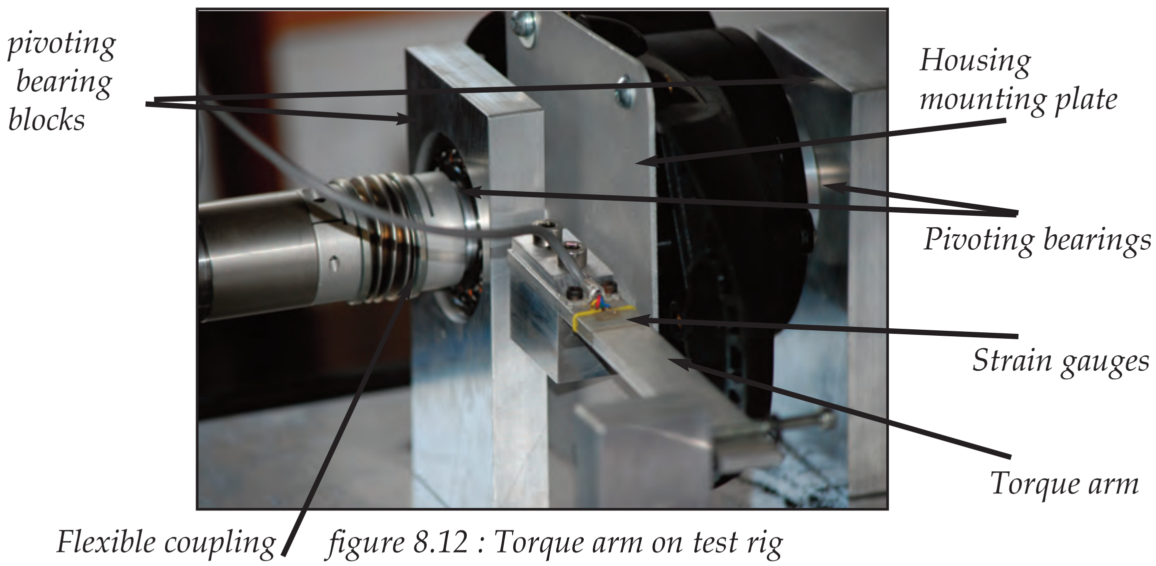


figure 8.12 : Torque arm on test rig

The torque arm has four gauges near its root detecting bending moments with the two gauges on the top surface going into tension and the two gauges on the bottom surface going into compression. These are connected into a full bridge the output from which was fed directly into the differential amplifier input of the data logger. A parallel trimming resistor was connected across the bridge so that the output could be zeroed. The torque arm was calibrated by hanging a 5 kg weight on the arm at a distance of 204 mm from the brake axis.

This applied a torque of exactly 10 Nm. The calibration process is shown in the figure 8.13.

figure 8.13 : Calibration photograph for brake test rig

The data logger has the facility to introduce scale factors so that the channel outputs can be made to indicate engineering units directly, the rotational speed channel being set to measure in rev/s the brake torque channel being set to measure in Nm.

The rotational speed was calibrated using a hand held tachometer as shown in the photograph figure 8.14.

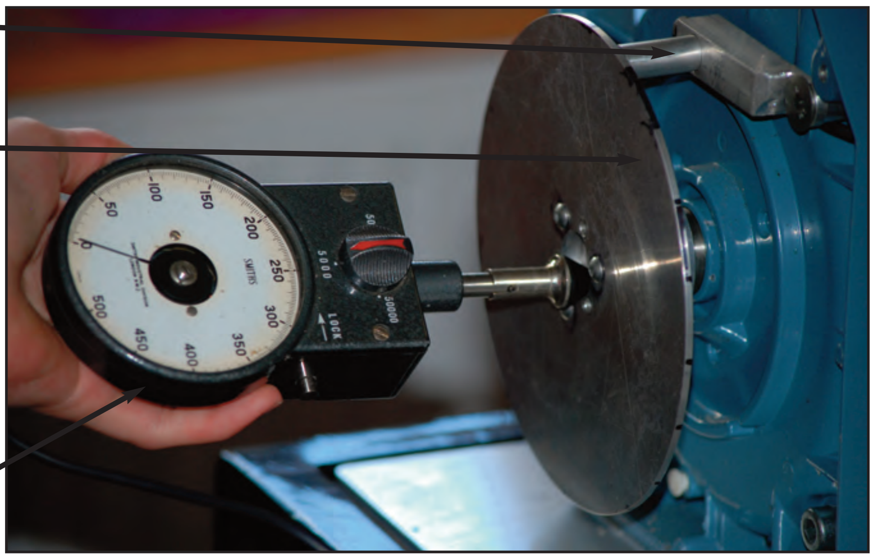


figure 8.14 :Tachometer calibration of torque test rig speed

8.6 Drop test rig instrumentation

The instrumentation connected to the drop testing rig was designed to measure the drop distance and drop speed when a weight is released and drops away from the unit pulling the tape off the internal drum. The photographs (figure 8.8 & figure 8.9) give a general view of the set up.

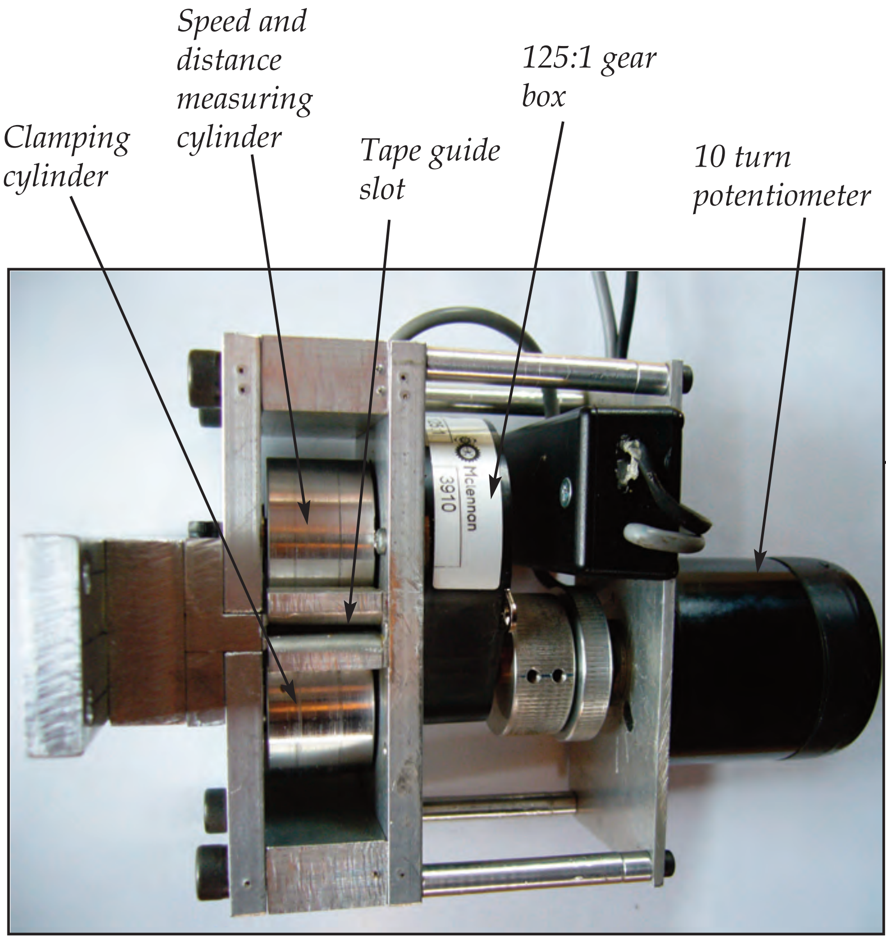


figure 8.15: Drop test instrumentation set up

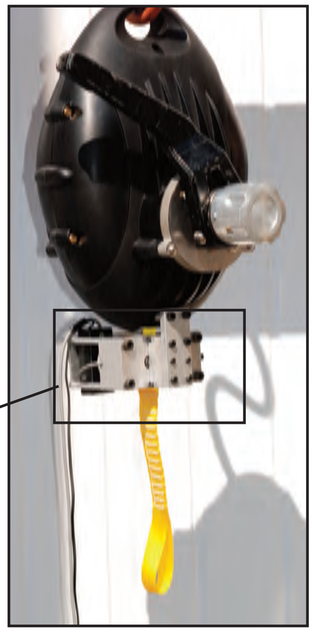


figure 8.16: Drop test rig on concept design

The tape passed between a pair of cylinders, one of which was fixed and the other spring loaded so that the tape was gently trapped between the two cylinders thus the cylinders rotate when the tape is moved between them.The cylinders were both

31.8 mm in diameter so that one cylinder revolution corresponds to a tape distance of 0.1 m. The shaft of the fixed cylinder was connected to a 125:1 reduction gear box and the output shaft from the gear box is connected to a 10 turn potentiometer so that a drop distance up to 125 m could be measured and recorded.
One face of the fixed cylinder was polished and painted with matt black strips and a small reflective optical switch was embedded in the mounting plate as shown in figure 8.9.

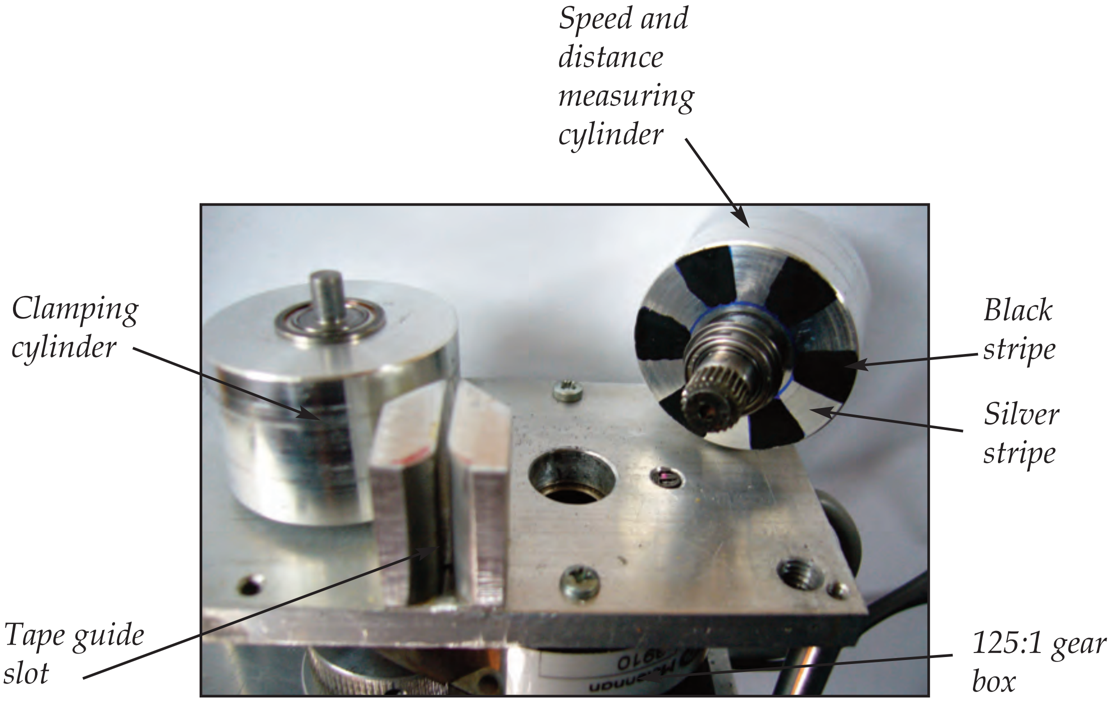


figure 8.17 : Pinch rollers shown exploded

This instrumentation was similar to that used in the brake test rig which used similar circuitry to that shown in figure 8.11.

The drop speed was calibrated using a small electric motor to drive the cylinder and a hand held tachometer that measures its rotational speed as shown in figure 8.18.

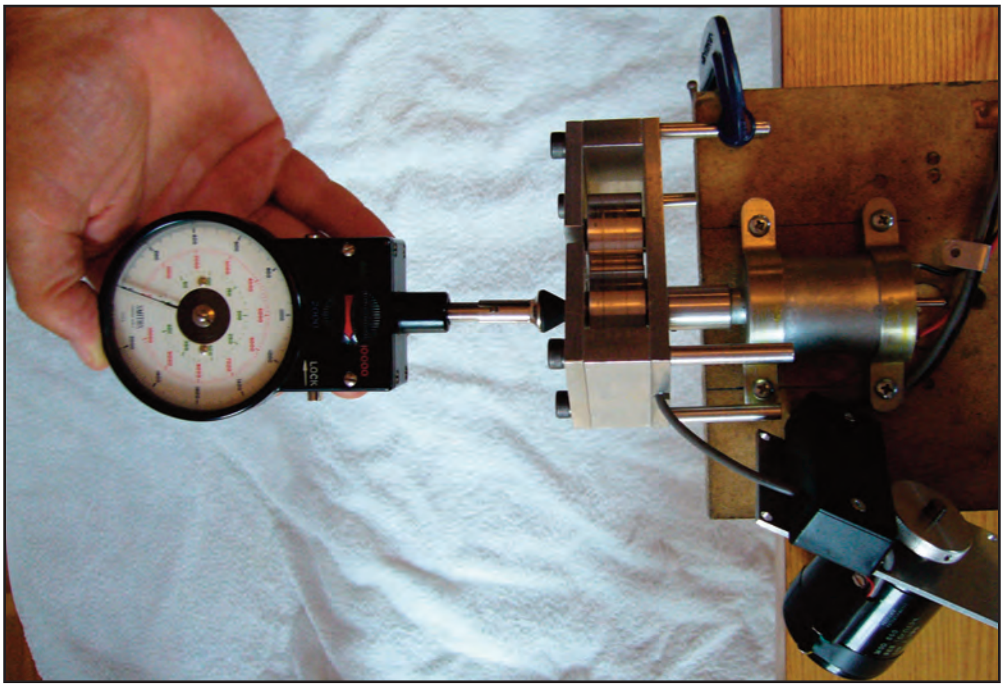


figure 8.18 : Tachometer check of drop test rig - calibration

As the circumference of the cylinder is 0.1 m, a rotational speed of 10 rev/s is equivalent to a drop speed of 1 m/s. Both the drop test rig and the brake torque rig outputs from the 10 turn potentiometer and the frequency to voltage converter are fed into the Pico high resolution data logger.
The drop distance channel was set to measure the drop distance with a resolution of 0.1 m and a range of 125 m. The drop speed channel is set to measure the drop speed in m/s with a resolution of 0.1 m/s and a range of up to 5 m/s.

8.7 Brake test using the torque rig

The prototype was assembled and mounted on the brake test rig which was used to run all the brake designs and to determine the torque generated for each configuration.
Data was recorded using the Pico data logger and was compared with the theoretical values.
After each test the rig was allowed to cool down to room temperature, also the brake band was checked to ensure that it remained located within the housing.The coupling that connected the input shaft of the brake to the output shaft of the motor had to be checked in order to ensure that the clamps were very secure, otherwise slippage was found to occur. The brake band construction weld was checked to ensure no weld metal or oval shaping was present as this could influence the test result.

8.7.1 Sliding 4 shoe brake

The sliding brake produced a torque of 16 kN and the curve follows the predicted quadratic polynomial curve accurately, the deviation from theory is considered due in main to the value of coefficient of friction used in the theoretical analysis. By adjusting the value of the coefficient of friction the theoretical curve follows the test results very closely.
The results for the power curve again follow the predicted cubic curve accurately up to the 3 kw power rating.The motor size on the test rig was 3 kw as this motor size was the largest allowable on a domestic 240 v supply. The tests were repeated several times and the brake performance gained a repeatable result suggesting that the variants are the coefficient of friction and the brake band condition. The coefficient given in the technical sheet for the material (appendix 3) is an approximation based on given mating materials used to run against. Upon consultation with the friction material company there was agreement that the value was an approximation and furthermore they have no way of accurately determining the figure.
It was, therefore, decided to run the test rig and adjust the theoretical coefficient of friction until the best correlation was achieved.
The resultant plots shown in figure 8.19 show a good correlation between the theoretical and test results. The brake torque data was fitted to a quadratic curve and the brake power was fitted to a cubic curve.

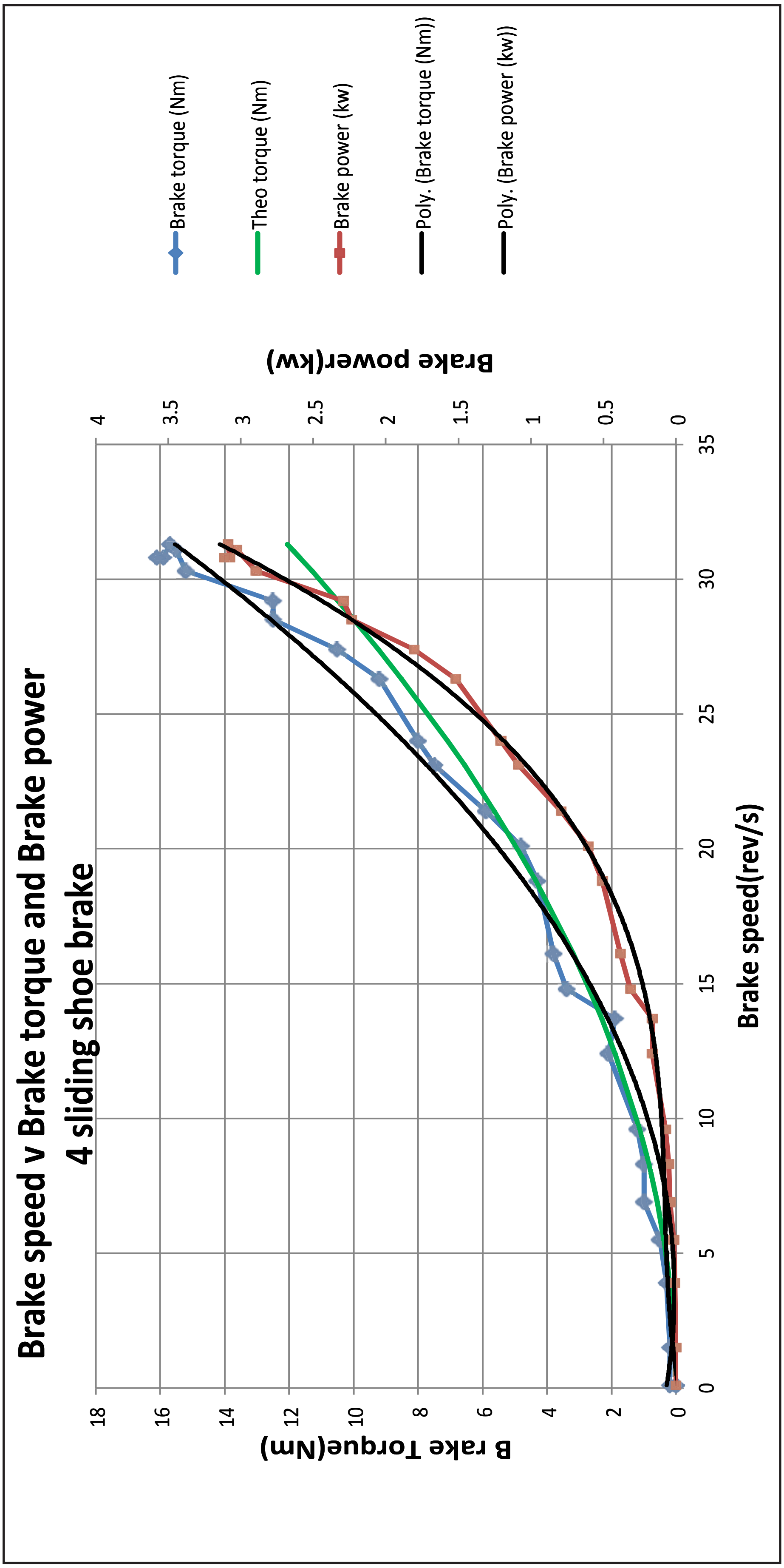


figure 8.19 : Brake speed against torque 4 sliding shoes

8.7.2 Pivoting 4 shoe brake

Following the initial tests on a pivoting four brake shoe configuration new brake shoes were manufactured with two pivot points at A1 and A2 (chapter 7 part 7.3) and a rebated slot to enable the shoes to move out without sticking. A series of tests conducted with both pivot points to investigate the reaction of the brake when the pivot point was moved, figure 8.20 plots the results from the test against the results obtained by theory.

From the plots the pivot point A2 produces the higher torque and although it follows the quadratic curve accurately its deviation from the theory as the brake speed increases above 15 rev/s is diverging, however, the curve of the graph line is similar. By using the shoes designed for pivot point A1 and machining a second pivot point at A2 the geometry of the brake has changed which influences the points of contact and force. The motor power of the test rig also has an influence as the higher torque generated by the brake is such that the motor is struggling to maintain its momentum without stalling.

The plot for pivot point A1 has a good correlation with theory suggesting that the geometry of the shoe is more consistent with the geometry of the brake band. It aso validates the theory and assumptions made as accurate.

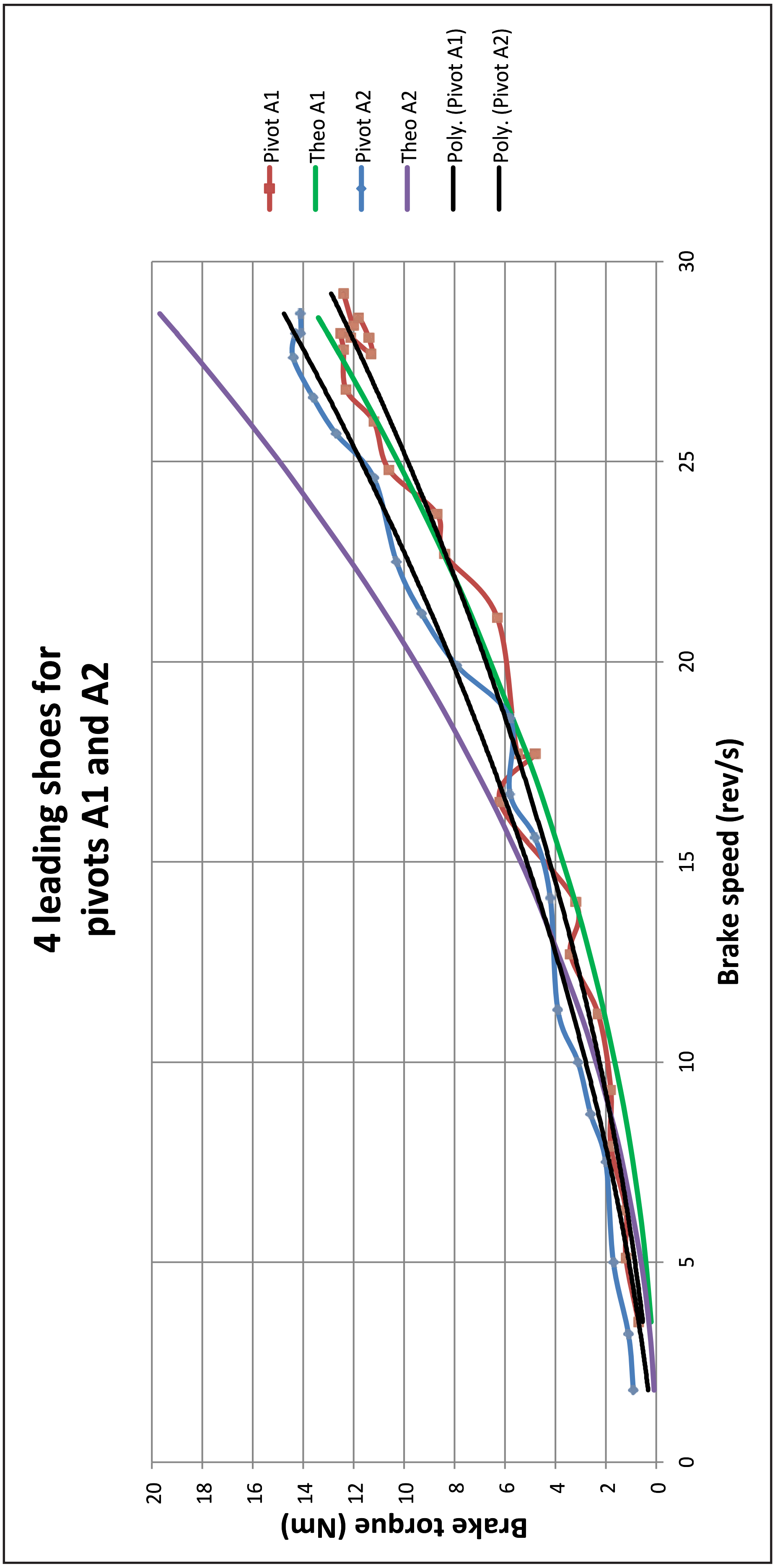


figure 8.20 : Brake speed against brake torque 4 shoes pivots A1 and A2

8.7.3 Pivoting 4 shoe brake against sliding shoe brake set up.

Figure 8.21 superimposes the plots achieved for the sliding brakes and the pivoting four shoe brakes, using the A_1 pivot point.
The four sliding shoes and the 2 leading + 2 trailing shoes give similar results on the test rig. The performance of both braking designs was repeatable. The shoes used in the pivoting design were lighter than the sliding shoes from which they were derived. There was no evident effect of stiction with the sliding shoes. The amount of movement of the sliding shoes is small in comparison to its spigot length and conversely the movement outward is small and supported from twist by the brake hub collar. Again the stiction is reduced due to the brass and stainless steel material combination.

Also included in the comparison is the 2 leading and 2 trailing brake results which produce lower torque than the other brakes but as with the sliding brake can operate in both directions which extends its use to reciprocating designs. The descent speed as noted in the tests with two shoe designs indicate that it can control the speed within the limits of 2 m/s, therefore its inclusion in reciprocating designs would be acceptable.

The figure also indicates that for the A2 pivot point the rig was on its limit for stalling the motor as the speed above 30 rev/s

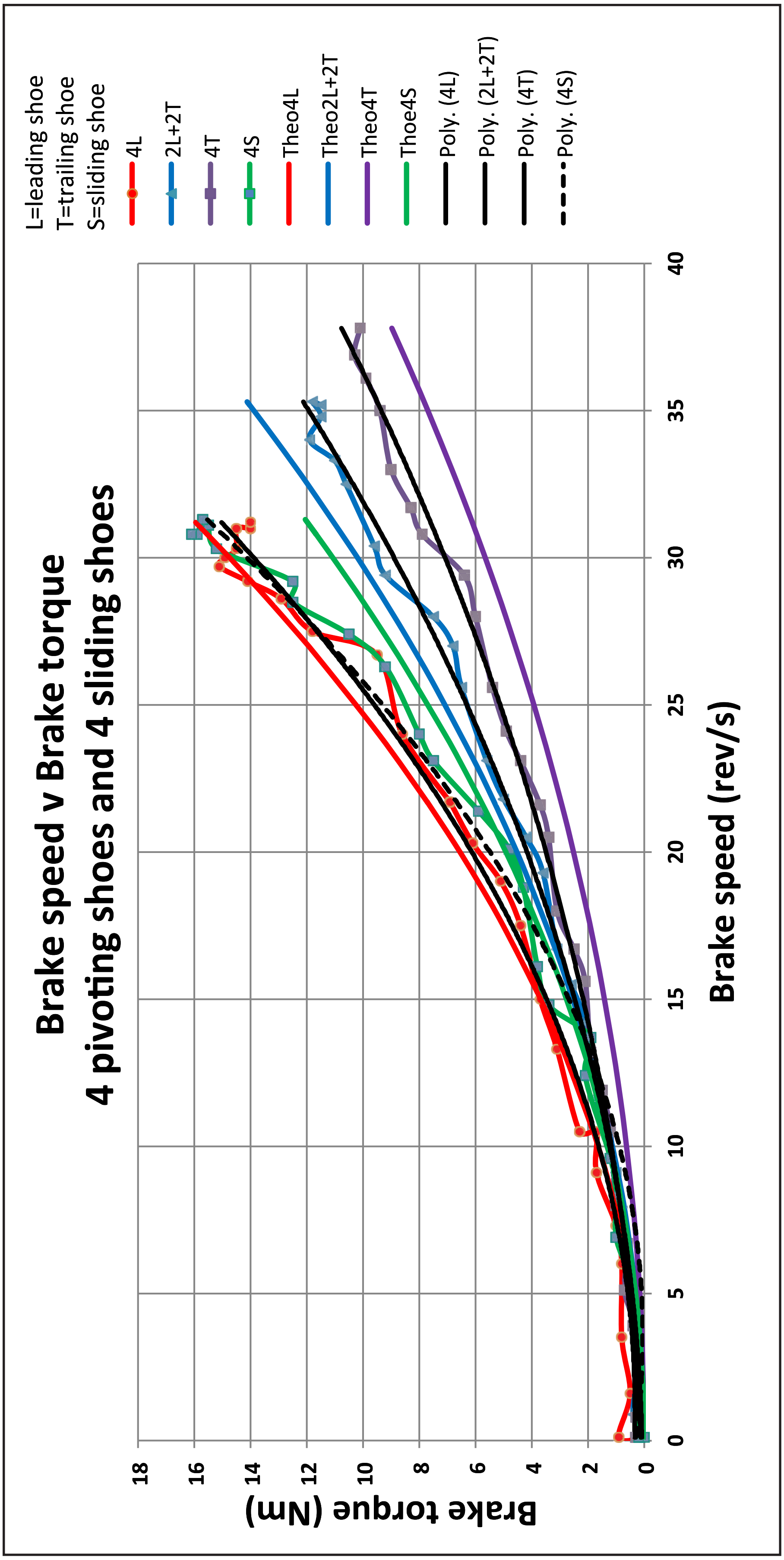


figure 8.21 : Brake speed against brake torque comparison 4 sliding shoes against 4 pivoting shoes

8.7.4 Pivoting 2 shoe brake

In figure 119 the plot of brake speed against torque shows that in comparison to the four shoe designs the amount of torque produced is substantially less at around 40 % of the other brakes. Not surprisingly the two leading shoes produce the highest torque of the three sets ups. In each case the torque produced correlates very accurately with the theory and quadratic plots. The torque is comparable to that produced by four pivoting trailing shoes and would suggest that with higher loads it would struggle to control the descent speed.

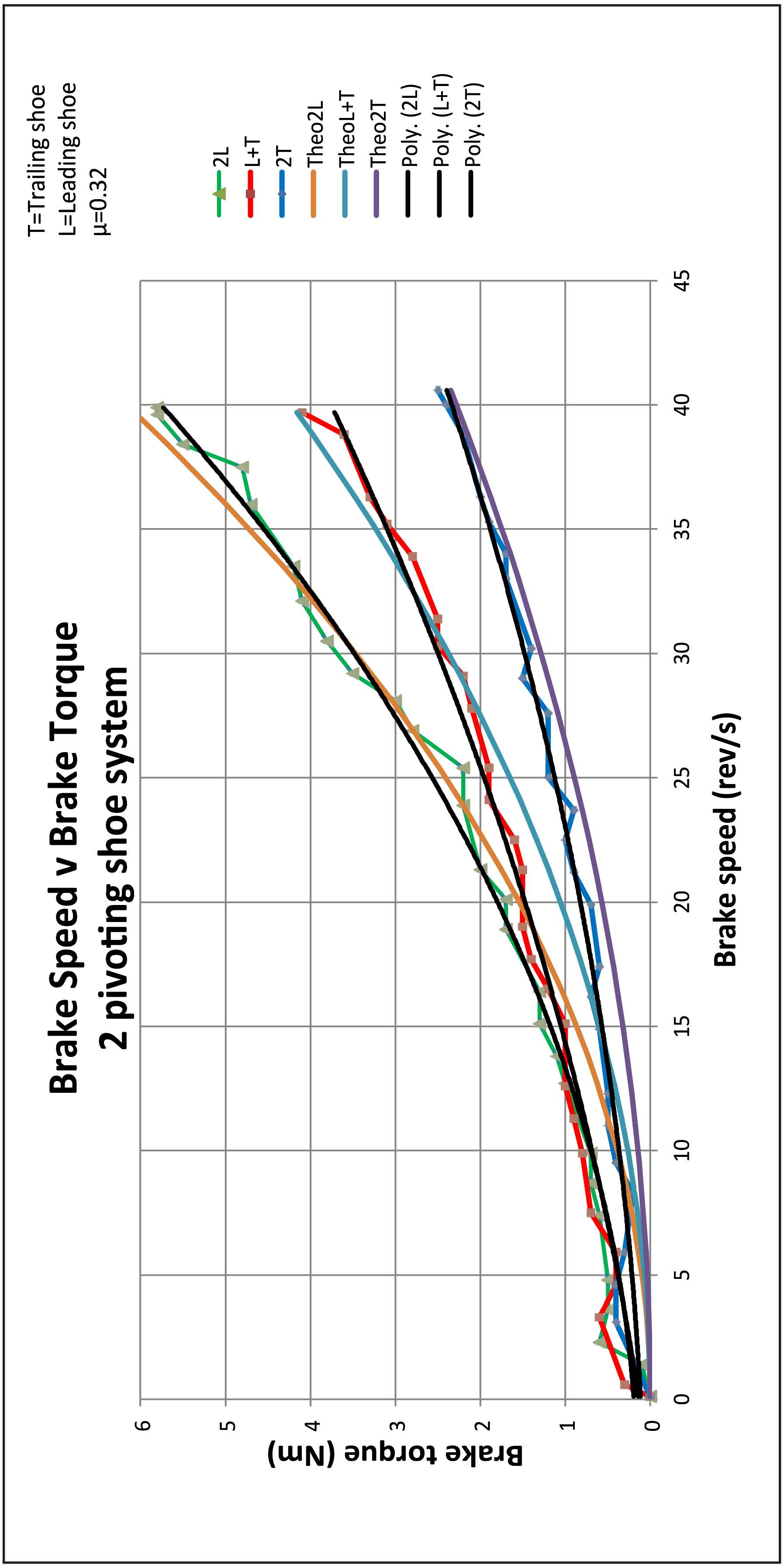


figure 8.22: Brake speed against brake torque - 2 pivoting shoes

8.8 Drop test using the drop test instrumentation.

The test was carried out at the National Lift Tower in Northampton figure 8.23. The tower has a maximum drop height of 105.36 m and is shown in figure 8.24. The lift car can be located at a number of floors where power is available to run the electronics. The descender with the rig was located at floor 11, the weight was suspended beneath the carfloor with the unit in the car suspended on an anchorage point in the roof of the car. A winch was used to take up any slack in the lifeline and a quick release was attached to the weight from the winch on release allowed the transfer of the weight directly to the descender. The various f loor levels available are shown in figure 8.24. There is the ability to inch the cab to any point but at the time of the tests the car had no internal power source. Additionally there is the safety implications of having all the people and test equipment inside the car.

figure 8.23 : National lift test tower

Floor	Distance from car floor to pit floor (mm)
1	6350
4	17880
5	28470
6	39290
8	49200
11	59470
13	72290
14	83250
15	94250
16	100250
17	105360

figure 8.24 :Floor layout digram - National lift test tower

The test was conducted using a 40 kg mass and in figure 122 the results have been plotted against the theoretical and measured quadratic polynomial for comparison based on an efficiency of 0.68 or 68%.

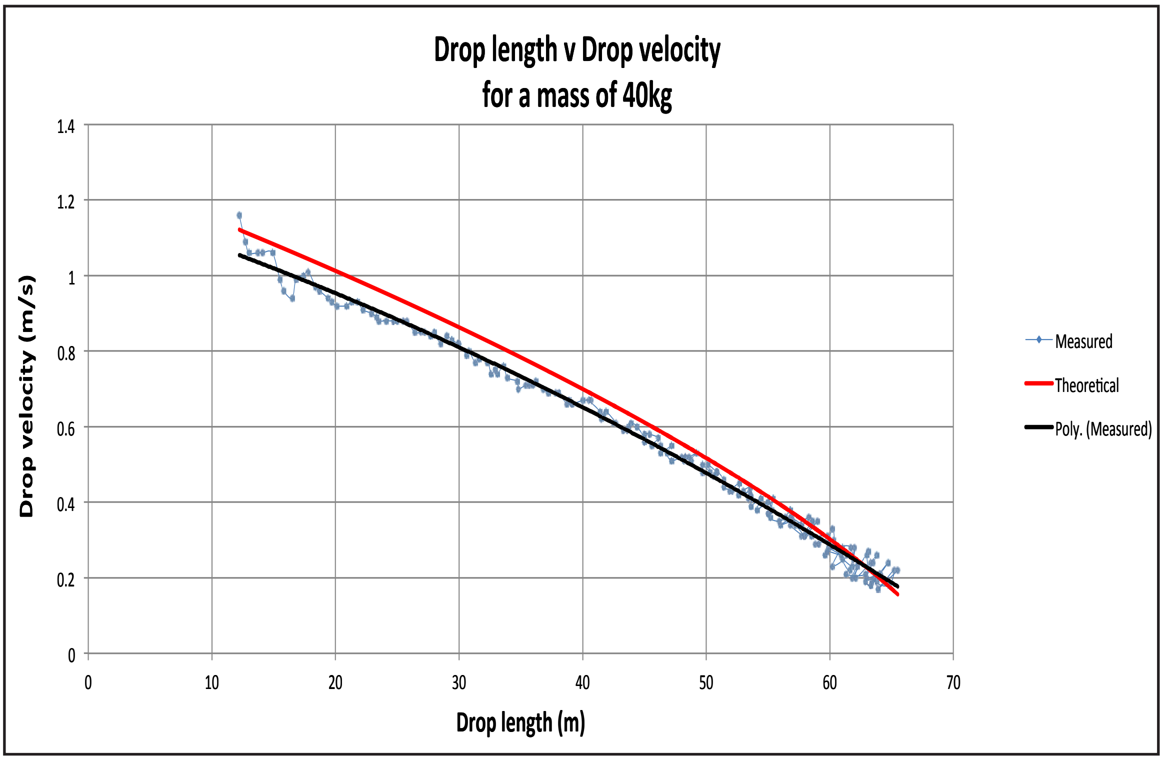


figure 8.25 : Drop length against drop velocity

One factor that was omitted previously in drop test analysis was the effect of starting the mass with zero velocity. In order to see if this is a significant factor the mass is considered with regard to the time and distance achieving 3 m/s with an acceleration of:

$$9.81\ m/s^2$$

$$v^2 = u^2 + 2as, \therefore\ s = \frac{v^2}{2a}\ ; s = \frac{9}{2\times 9.81} = 0.45\ \mathrm{m}$$

$$v = u + at\ , \therefore\ t = \frac{v}{a}\ ; t = \frac{3}{9.81} = 0.3\ s$$

Compared with the distance of 60 m and the time of 60 s both of the factors are very small and can be ignored.

Neither the coefficient of friction nor the dynamic transmission efficiency are known very accurately and it can be seen that a small reduction in the theoretical values of either or both of these would cause the theoretical curve to closely follow the measured data.

8.9 Drop test example Liverpool sliding brake

In order to carry out dynamic tests at height the EON coal terminal in Liverpool was set up to carry out repetitive tests over 44 m, the temperature of the casing was also recorded to determine the maximum temperature of the surface following 16 descents using a test mass of 75 kg in accordance with EN 341:1992 figure 123. The weights were calibrated prior to the test by suspending them from a certified load cell to ensure the weight was within accepted limits of +/- 1kg, the weight was stated as 75.1 kg within required limits.
The time was recorded for each successive drop and the results are shown in figure 124.

figure 8.26 : Brake test set up Liverpool - sliding brake

The weight was suspended from a hoist using a quick release hook the weight was released triggering the timing of the descent, at the end of each descent the weight is raised and then released again, there was no requirement to pull through the rope as the unit used was reciprocating with anchorage points for the weight on each end of the rope, this kept the time between drops to less than 1 minute. The temperature rise was recorded for the period and the peak temperature recorded on the housing surface was 42.8 C, the International limit for temperature is 48 C. Due to the nature of the test set up the prototype was subjected to shock loading as despite the brake being permanently engaged there was an element of free fall due to the initialposition of the weight and the quick release mechanism. The four brake shoe sliding configuration was used for the testing.

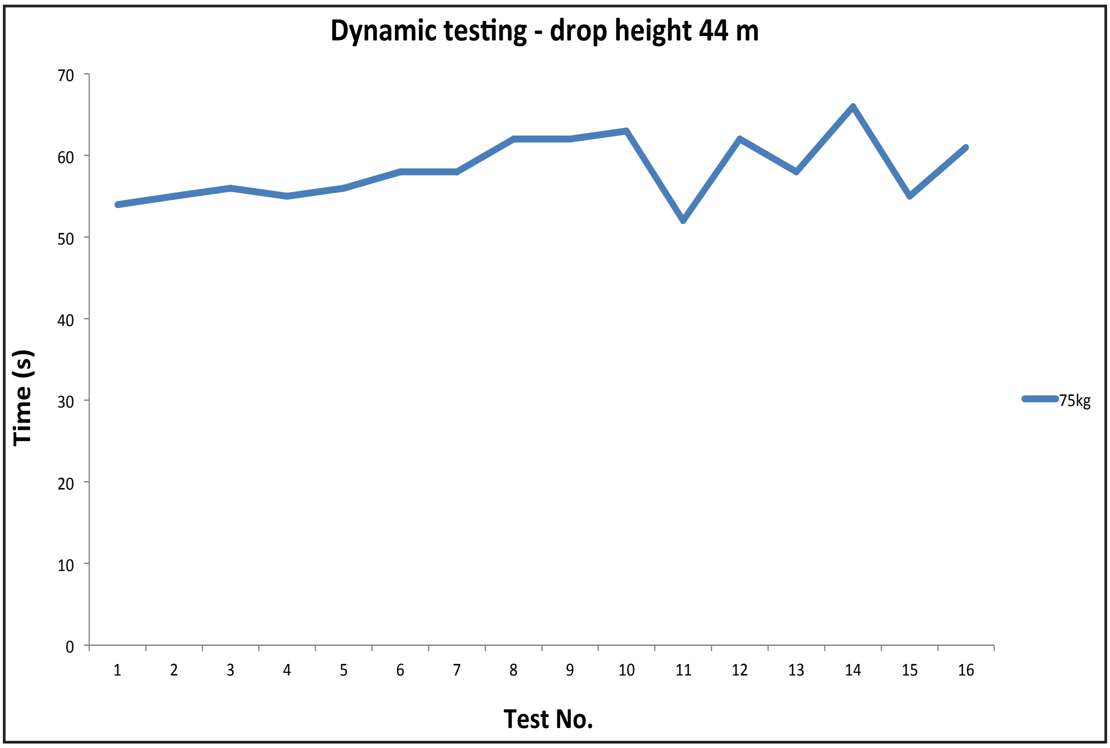


figure 8.27 : Test number against time

In the figure 8.27 the plot of test against descent speed is given, the descent speed should not exceed 2.0 m/s, all the tests conducted produced a descent speed average of 0.76 m/s and peak of 0.85 m/s.

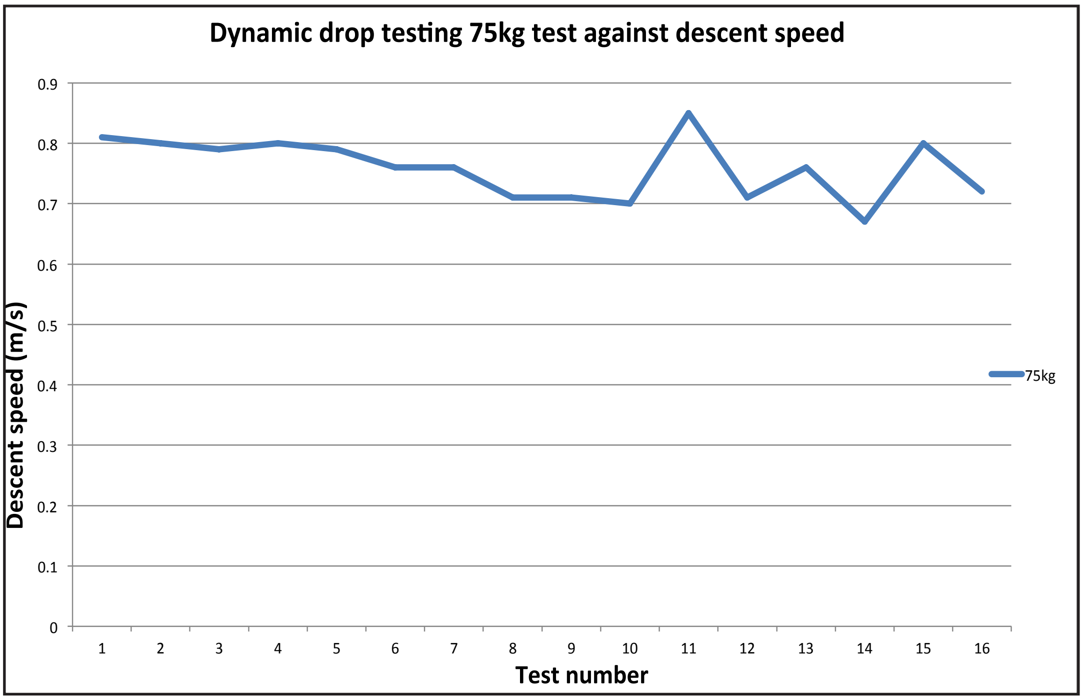


figure 8.28: Test number against descent speed

8.10 Dynamic and static testing carried out at SATRA

Dynamic tests and static tests were carried out at the SATRA technology centre where the prototype was dynamically tested using 30 kg and 150 kg test weights. The objective of the tests was to see if at the two extreme weights the brake can without assistance control the speed to less than 2 m/s. The tests concluded that the sliding brake met the requirement (appendix 2). The tests were conducted with the rope both dry and wet to see if dampness affected the result. The test tower was 10 m in height and each test was repeated 5 times to ensure repeatability. The weight was released by a switch at ground level. Figures 8.29 and 8.30 show the dynamic tower.

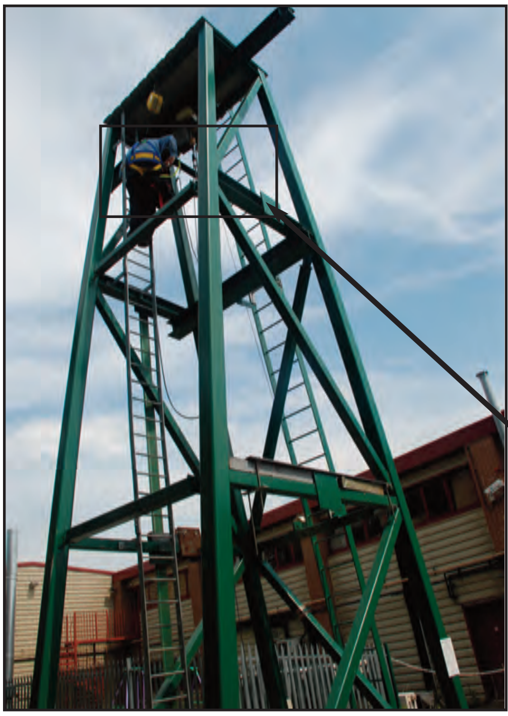


figure 8.29 : SATRA external dynamic test tower

figure 8.30 : Close up of the prototype descent unit prior to testing

All tests were conducted using the four shoe sliding brake inside a 1270 prototype unit.

The 150 kg and 30 kg tests were performed satisfactorily and the static strength of 12 kN was maintained for 3 minutes and then the unit was taken to destruction, failure of unit 1 with fracture of the handle at 14.2 kN and unit 2 fracture of the handle at 15.9 kN (appendix 2)

The handle was completely constructed in glass reinforced nylon without any metal inserts.

8.11 Dynamic test rig

In order to carry out dynamic tests on the prototypes a dynamic test rig was constructed.
The total drop height was 7.5 m with a beam set at 5.5 m, there were three fixed anchor points on the main beam and additional anchorage points was achieved using girder clamps as shown in figure 129 . All forces were recorded by using a HBM data acquisition logger and "Z" beam load cells rated at 20 kN. Staging was provided to enable the set up at either the beam or top level anchorage point. Test weights 30 kg, 100 kg and 150 kg were manufactured to EN 364 and together with the rescue dummies and articulating manikin performance drops were performed and used to observe the motion.

figure 8.31: Dynamic test rig designed to carry out function tests with weights up to 150 kg

figure 8.32 : Close up of the prototype descent unit after testing on the rig with tape evident from the unit

8.12 Static test rig

The prototypes and the lifelines required static testing to 12 kN. A test rig was constructed with a 2 m bed and a 1 m hydraulic ram. A load cell and electronic peak force recorder were used to record the result of the test. Also the HBM data acquisition logger was used to record the complete static test directly to computer for analysis later. The test piece was located at the fixed end under a window guard so that it was observed during testing without risk. The test rig was tested to 20 kN to ensure that the power pack and setup were capable of achieving a minimum of 15 kN which is the limit set by EN364 and EN 360 for synthetic lifelines which use tape or webbing, this goes beyond the descender requirement of 12 kN

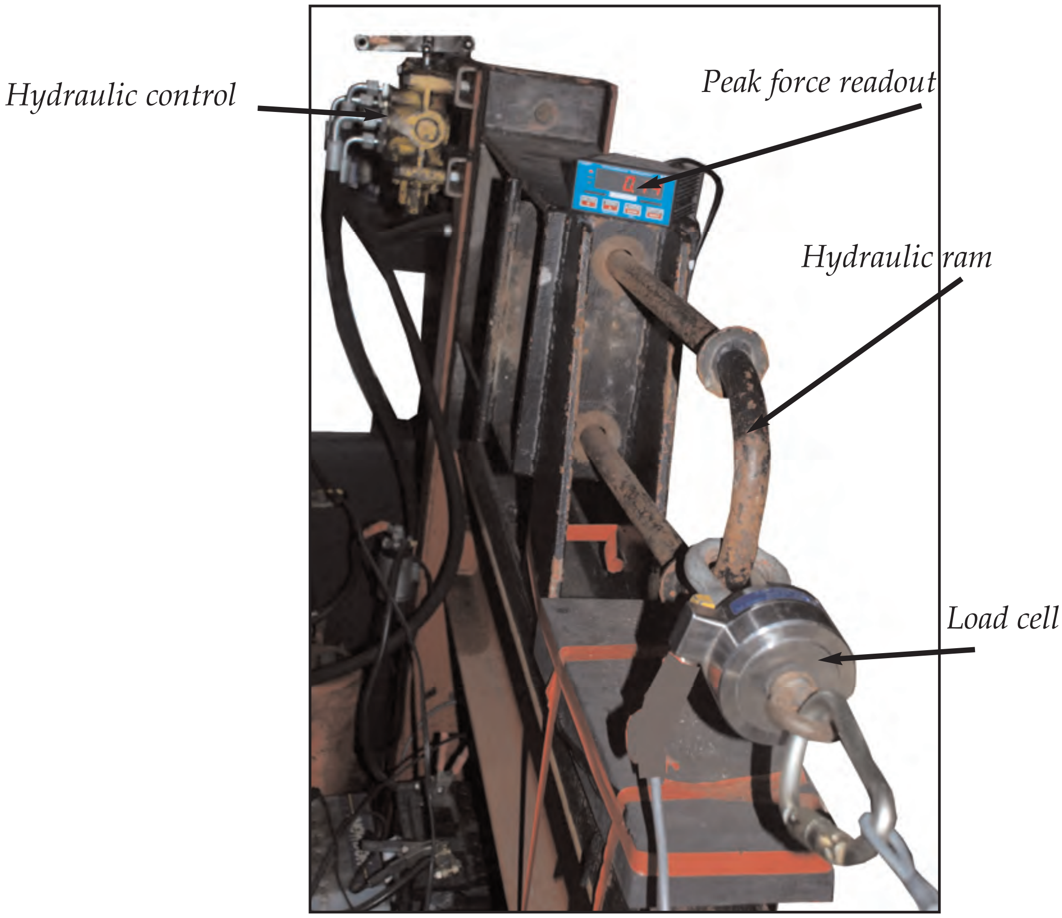


figure 8.33 : Close up of the prototype descent unit after testing on the rig with tape evident from the unit

8.13 Discussion and summary

In this chapter the design brakes were tested using a number of purpose built test rigs and fixtures in order to prove the designs and compare them with theory. The presentation includes the additional considerations such as friction over the entry guides and determines the affect on the designs. The movement of the pivot points has been looked at and the correlation with both theory and between designs explored. It was shown that with the pivot points at A1 and A2 the highest torque was produced with the torque produced by the sliding and two leading two trailing configurations almost equal.
The trailing shoe arrangement with pivot points at A1 and A2 produced an almost match indicating that the pivot position is less significant for the trailing shoe design.
The combination of actual dynamic drops to that of using a bench rig has been developed to provide results that are representative and informative.

Chapter 9 : Conclusions and future work

9.1 Conclusions

A number of designs have been developed as part of this research. The designs have been validated by mathematical analysis, prototyping and testing. What has emerged is the ability to utilize a number of descent options depending upon the environment and application. The use of polymers such as Grivory for the casing and drum provides a lightweight alternative to aluminium and steel with improved environmental performance.

The developed Kevlar tape offers a new solution for the compactness of a design, and offers the ability to control descent speeds very easily. Objectives 1 and 6 in chapter one, relate to the development of a lifeline and the effects on descent due to gearing. The developed tape manufactured in Kevlar, or if temperature is not an issue with Dyneema, presents a solution to both of these objectives.This has implications in rapid evacuation from high rise structures.

The choice of the lifeline requires correct termination. The combination of the Kevlar tape with Zylon thread produced high tensile strength, which was about twice that of the other material combinations.

With a thickness of less than 1 mm the tape is able to provide a long descent from a small cartridge. The Kevlar tape has a decomposition temperature of over 600 ^{0}C with a high resistance to cutting or blow out flames. However, the use of any Kevlar or Zylon products should not be over exposed to sunlight in the descender as its mechanical properties would be affected. Therefore the tape would require protection from light until the moment of deployment and then discarded.

The simple box and gate sewing pattern provided the best solution to the termination of the tape as it managed to capture many of the longitudinal strands without damaging the weave.

By comparing the weave patterns produced, the herringbone pattern produced the best results during all the tensile tests. It is also an easier tape to sew and less prone to twisting due to the multiple bands within the construction.

Kevlar thread proved to be a poor option as the thread in shear simply failed at very low tensile levels. The use of either Kevlar or Zylon thread does present additional problems of manufacture, as both are hard to cut on automatic sewing machines. They also have to be stored in darkness until required to prevent the thread being exposed to daylight.

Rope and wire, unlike the tape, have a role where repeated exposure and use is required. Wire and rope have practical advantage with descenders that are to be used continuously for example on a climbing wall, or mass evacuation in a disaster mode. However, one of the disadvantages of wire rope is its inflexibility, mass and minimum bend radius.

Fibre rope requires a diameter of 9 mm in order to meet the 12 kN tensile strength requirement as laid down in the standards. It also requires a large storage bag or similar, which adds weight thus restricting its application for portability and use. Rope also tends to knot and tangle easily which, in an evacuation could result in failure.

With all lifelines developed and tested the function is reliant upon the descent mechanism which has to be robust and reliable. As most descent equipment is envisaged to be deployed without the luxury of a back-up, reliability is paramount in order to preserve and save lives.

The brake designs developed as part of this research produced test results that compared closely with the theoretical analysis.The pivoting shoe brake with 4 leading shoes, produced the highest torque. The arrangement with two leading and two trailing shoes produced a similar torque to the four sliding shoe brake, although the shoe mass was considerably less.

The two pivoting shoe design produced the lowest torque and unless gearing was significantly increased the two shoe design would struggle to perform the primary function of controlled descent. The 4 pivoting trailing shoe brake produced a low braking torque and the pivot point was not critical.

By adjusting the theoretical coefficient of friction or the gearing efficiency for the brakes, the theoretical and practical results are almost the same. By experimentation it is now possible to estimate the coefficient of friction for any given combination of materials, by adjusting the theoretical curve to be coincident with the practical test results. This is a useful addition considering the approximations made by friction material suppliers and enables better prediction of performance and hence design.

Considering the results obtained from the tape drop test where the radius of the tape on the drum reduces from its initial radius to its final radius, it can be seen that by using the different radii the design can be modified such that the initial descent speed is greater than current standards permit but the ground contact speed of 2 m/s is achieved within those same standards. However, in the way in which current standards are written it may prove difficult to achieve approval of a design with a varying descent speed higher than the 2 m/s.

The drop tests performed at the national lift tower demonstrated that the gearing and the descent speed reduced as the webbing paid out as predicted, offering the potential to use spooling where by higher descent speeds at the initial stages can be used to descend from greater heights in shorter time periods as set out in the objectives.

All the brake tests when plotted were found to follow the cubic and quadratic curves proportional to N cubed and N squared for the power and torque respectively, as predicted by the theory.

This research offers a reference for engineers and designers, in order to design descent controllers for a wide range of applications and conditions. There has been a dearth of information and research into descenders and how they function, with information difficult to find or not available at all. What this research has achieved is to provide an insight into the design of descent devices that previously had not been addressed.

By developing test rigs that give predictable and repeatable results, tests required by International standards can now be performed.

This research has contributed to knowledge, in respect of the design factors such as torque, power and brake friction and the use of descent control devices. It also shows how devices can be tested. This contributes to the education of engineers and the provision of data helping to make descent control safer and provide information to help rectify the dearth of information on the subject.

9.2 : Future work

Personal evacuation and mass evacuation systems are predicted to be a fundamental consideration for buildings and structures in the future. This is due in part to the demand for more high rise accommodation and work areas caused by land shortage and cost implications. This is driving the need for more solutions to be developed for descent systems. In addition, the advent of high profile terrorism, now considered a fact of modern living, will move forward the use and research of descent devices and systems as engineers of the future work on the problems associated with evacuation from height.

The next stage in the development of the personal descent controller and the basis of future work is to investigate how mass evacuation is achieved using the brake mechanisms and lifelines developed here, including the use of cartridges that can be used to effect multiple descents. Future work would also include alternative braking methods with the descent controlled by hydraulics or pneumatics.

Work on the position of strong anchorage points on the structure for evacuation, in addition to the use of lift shafts or blow out windows would form part of future work. This would integrate descenders with other equipment such as smoke hoods or guided escape methods. This work would allow for the inclusion in new technology, which presents new demands, such as wind farms, leading to an expanded scope for future work.

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Appendix 1 - evacuation device

Appendix 1 Descent control review

A1.0 Introduction

The principal behind egress methods is to provide a means of emergency escape for any person whilst working or residing at height. In the event of an incident then conventional methods of escape may not be possible. Lifts would become static, ladders and walkways potentially unsafe with fire escapes involving numerous landings and levels impracticable or impassable. If one then add the effects of smoke, heat, fire, gases, structural collapse or chemicals then entrapment is a real possibility. Disabled or injured persons would also present a challenge, the challenge being to get them down safely.

When one looks at modern structures, industrial or private and read the contingency information provided there is a real lack of escape provision in the event of fire or structural collapse. The twin tower disaster had both these factors demonstrated in the harshest fashion and with Piper Alpha you had fire as the principal issue. What was present in both cases was the lack of provision for escape from height where the traditional exits were blocked or the height and time were prohibitive.

In America one has OSHA regulations, which touch upon the subject. OSHA 1910.35 allows for the consideration of windows or other wall exits as a potential area for evacuation using descent devices.

When considering wall exits, irrespective of type chosen the first consideration has to be where one connects a descent device (the anchorage) and how one connects it. Even before an exit is chosen the anchorage has to be present and clearly identified. It is possible that the equipment would provide an indication of temporary anchorages that would work.

It may be impracticable to spend time searching for a marked and approved anchorage. The collapse of the twin towers demonstrated the relative short time frame that one may be presented with, in any event the rule is clear and that is a person must be get away from the problem as quickly as possible.

There has to be a compromise between many factors but EN341-1993 states that the descent speed is limited to 2 m/s and strength is 12 kN. If these are to be applied to high level evacuation then they may prevent a useable solution due to size and time constraints. The 12kN level was introduced to equal the fall arrest standards such as EN360-2002 where lifelines have to manage shock loads. However in the case of descent controllers the argument is that they will pay out and most are continuous so the shock loadings developed are minimal. In most if not all cases tests have shown that the peak forces seldom go above 10% above a persons weight. This would give a level of 2 kN as a requirement for line strength and would allow smaller gauges.

In parachute terms a parachutist who employs a body roll landing would be capable of 6.4 m/s (NAS804) and if one was evacuating to water then 4 m/s may be more practicable, even on land 4 m/s may be a more useable speed of descent. In developing a device variable rates may be applied or even zoned evacuation, for example one may supply devices to floor levels which operate a different speeds and may slow as descent distances reduce.

A recommendation by Harold Steinberg in his 1977"Study of Fall Safety equipment" concluded that 15 ft/sec or less for uninjured persons and 10 ft/sec for injured persons be adopted. In the event as demonstrated by EN341-1993, 2 m/s or 7 ft/s was adopted.

A1.1 Vertical descent

Escapes from Oil/Gas production platforms, overhead cranes, hotels, office blocks, chimney stacks, and elevated workstations are examples of typical applications for vertical descent. Vertical systems are generally designed to allow the descent of one or two persons at any one time per system. The design of some devices is that once the first person is down then a second may descend, like a see saw arrangement, (reciprocating) figure A1.2

figure A1.1- single person only

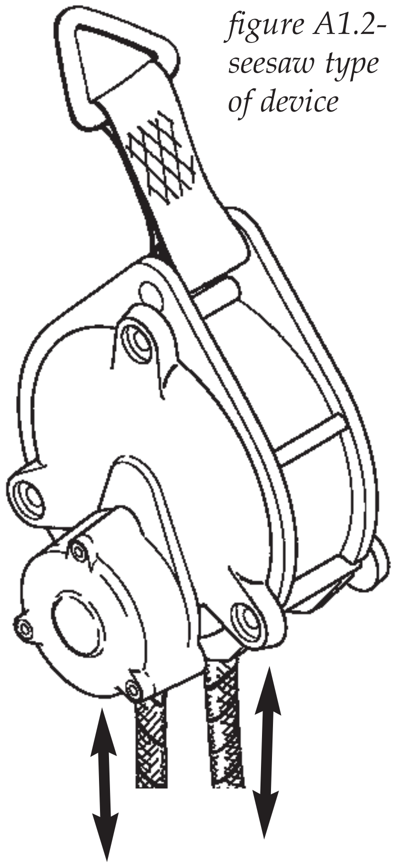


figure A1.2- seesaw type of device

Descent controllers fall into two categories irrespective of the method employed. They are either automatic or manual. An automatic descent device is one which is pre-set with a descent speed and cannot be adjusted or over ridden in use. A manual device is one which requires user control at all times (figure 130).

Automatic devices operate at 2.0 m/s +/- 10% in line with standards and the longest device found on the market has a distance of 35 m. The weight of lifeline or rewind unit size being major limiting factors.

Manual descenders have to operate within the same speed constraints but it may be possible to increase descent distances to greater than 60 m. For practicable reasons they would normally be less than 100 m. If a pulley type systems is used then 20 m would be more normal as they work on either a 4 or 3:1 ratio which would require substantial amounts of rope and also run the risk of ropes being tangled.

A1.2 Angled descent

If a structure is collapsing or if a fire is possible at the base of the structure then angled escape is considered. The method normally employs a guideline in order that the user can escape. As a rule the speed of descent would be greater but the removal of a user from the danger zone is considered essential, so this lends itself to a variable descent speed which is considered in the main body of the research.

A1.3 Descent devices

A1.3.1 Friction

These devices can be manual or automatic or a cross between the two. They represent around 99% of all devices on the market and run on webbing, rope or wire.

The first common feature between all devices is that they require the user to wear an attachment point in the form of a full body harness or helicopter strop. For young children or injured/disabled persons the harness/strop can be substituted by a sling or seat device.

The second common feature is that they all have the same operating parameters that is a descent speed of 2.0 m/s and weight limits of 30 to 150 kg. In reality most devices are designed to take two persons descending together in order to allow for rescue so a more usual upper limit of 200 kg is applied for this design. The research aim is 150 kg. The higher level of 200 kg has been chosen to permit or allow for a rescue person and their equipment.

The third common feature that has to apply is the device must be compact, easy to use and light and transportable. Figures A1.3 and A1.5 show automatic descenders under trial as developed by the author of this research. Figure A1.4 shows a sketch of a traditional body device as used by helicopter rescue services for winching people onto their craft.

figure A1.3 - Evacuation with helicopter strop under test into water

figure A1.4-Helicopter strop

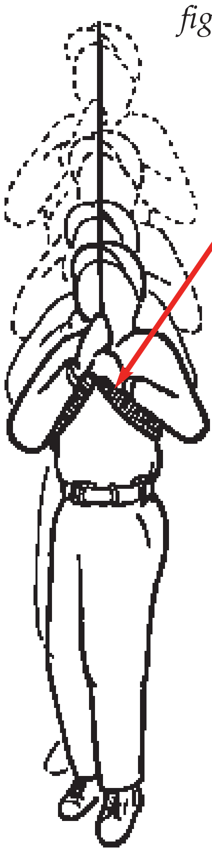


figure A1.5- Evacuation with full body harness under test from a crane cab

A1.3.2 Inertia reel

This type of device has a lifeline on a drum or spool. Lifelines can be made from wire, webbing or fiber (rope). As the user moves away from the device the lifeline pays out. To descend there are two main methods, the first has a braking mechanism that is permanently engaged and the other relies on a trigger such as centrifugal force caused by a fall which engages a brake.

The device drum has a spring that rewinds the lifeline when the user moves towards the device or disconnects from it. This type of device is restricted by the size of the rewind springs to less than 60 m and the rewind mechanism can have inherent safety issues at it rewinds the lifeline.

This type of device is fully automatic as the speed is pre-set at the point of manufacture and there is no provision to override the device once it is in action. If one is descending on this type of unit and a hazard occurs below, you such as a fire then there is nothing the user can do to overt the danger.

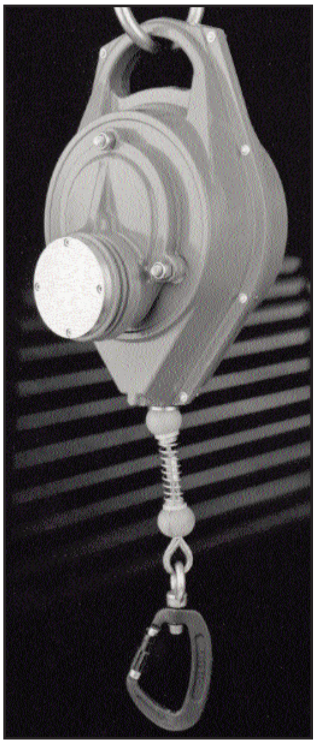


figure A1.6- An example of inertia reel prototype descender developed by the author

A1.3.3 Fall induced inertia reel descender.

This type of descender was designed as it offers descent only when a fall has occurred, this permits the easy rewind of the device in order that further persons can be evacuated. The problem is that the rewind in the unit was severe and there is the possibility that the hook coming back without control could injure an unwary bystander.

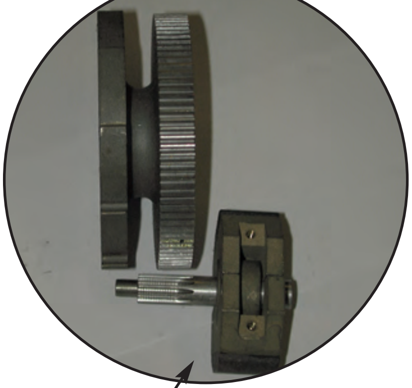


figure A1.7 side view showing the pinion and crown gear arrangement. The brake shoes shown are activated when the ring is brought into play by brake pawls that are located on the drum. The acceleration of the drum caused by a fall activates the pawls.

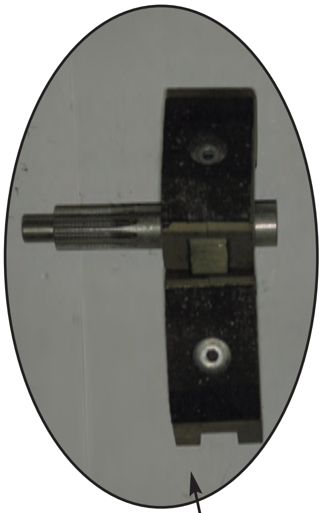


figure A1.8- Brake shoe arrangement

figure A1.9 - Top view of the brake mechanism.

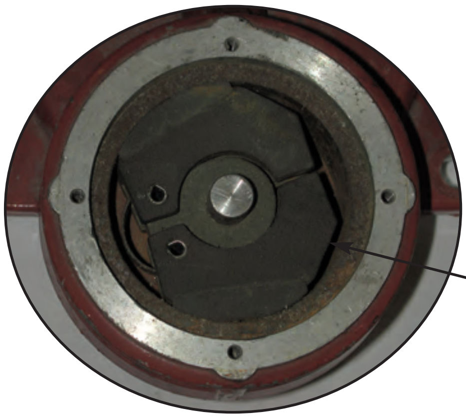


figure A1.10- Top view of the brake mechanism located in the housing. Speed variation can be obtained with the use of different liners and brake facings

Two versions of the device were made one with 20 m and the other with 35 m of wire rope

In addition samples were put into an adverse marine environment to see how they performed.

The housing was LM25TF (appendix3)with a liner manufactured from low grade stainless steel tube.

A1.3.4 Permanent inertia reel descender.

Following on from the fall induced descender the next step was to evaluate it against a descender which is permanently engaged. This would provide a controlled rewind. One question would be the effect on the rewind spring. In practice, due to weight considerations rewind springs are designed to operate at their limit. As a base design criteria the spring would have to rewind 1 kg +/- 20% at full out (full out means that all the wire is extracted under tension from the unit and coiled up the test weights are then applied to the coiled wire)and last 1 m, the force required to overcome the brake may be a problem.As with the previous type two models were manufactured up to 35 m.
There is a device manufactured in France which utilises a geared rewind action and that device is capable of 60 m (inertia reel only) using a 4 mm wire rope making it the longest inertia descender in the world. The design is extremely heavy and is not portable.The spring and gearing are very heavy using chain drives to assist the rewind, the unit was not very successful.

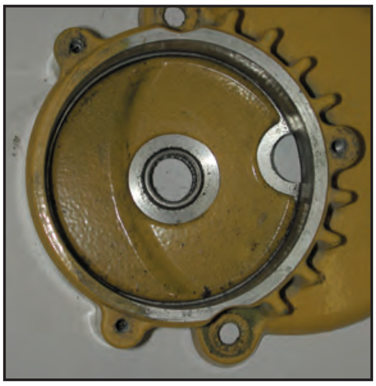


figure A1.11 - brake housing

figure A1.12- brake hub

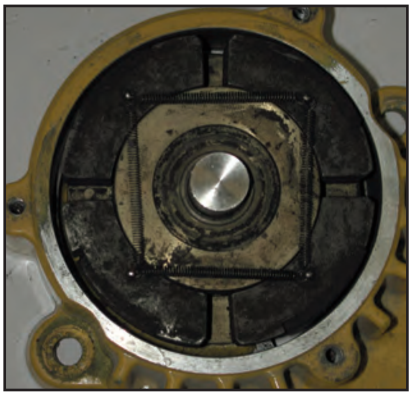
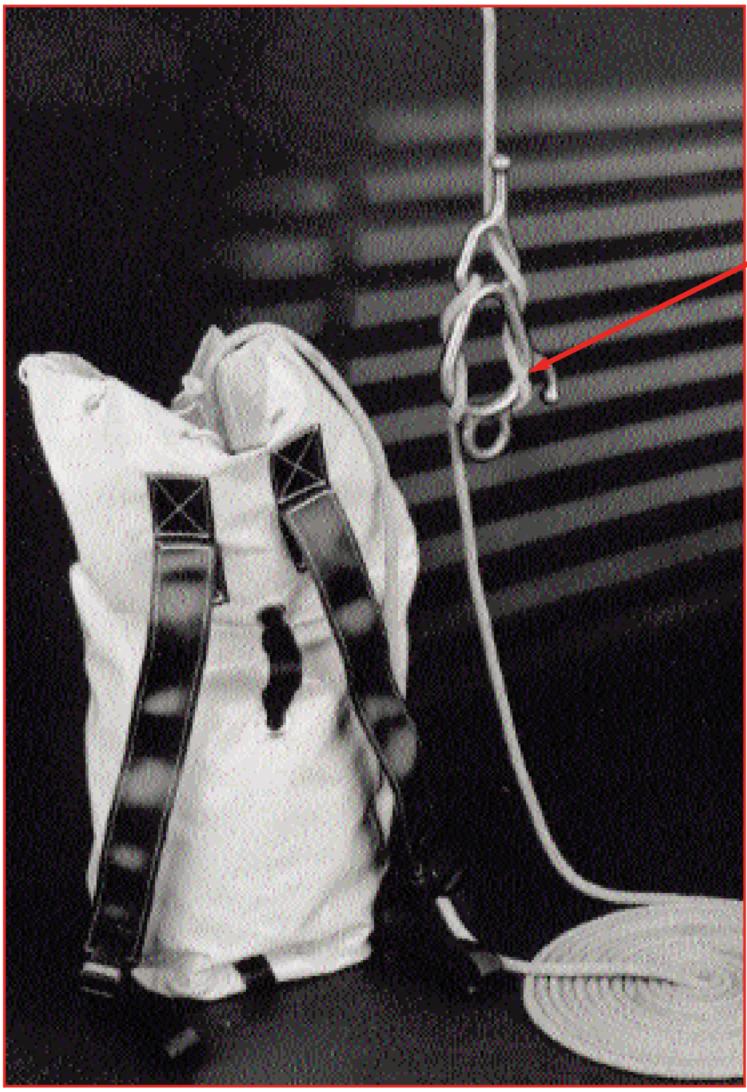


figure A1.13- brake assembly in housing

The design developed has a gear ring directly bolted to the drum, the drive gear is permanently engaged in the ring. In this case there are four brake shoes, one in each quadrant. The shoes simply slide in the hub and springs are added to retard the brake shoe movement for speed control.In order to alter the speed tests can be carried out using different liners and brake facings.

A1.3.5 Climbing figure of 8

Used in climbing for many years as an abseil device it relies on a rope passing around a forged frame, normally aluminium or titanium. The traditional shape is that of a figure 8, there have been a number of iterations adding horns etc., but the principal is the same in each case.
These devices are manual and rely on training for safe use they also put a twist into the fiber rope that reduces the rope life. Normal rope size is 11mm diameter, which does affect the practical use, as the rope becomes very heavy and hard to manage.



figureA1.14-Fig 8 example with lock off horns. Sample was made from titanium known as a "Fisk" Descender

A1.3.6 Spiral

This type of device is a development on from the figure of 8 abseil unit in that instead of twisting the rope around the 8 frame it is passed around a centre stem, the more turns the slower the descent. Unlike its forerunner this type of device can be produced with a manual override mechanism.

The override versions have the centre spindle and the outer tube. However the outer tube is sprung loaded so that if one applies too much down force one traps the rope and stops,if one lets go then the spring pushes the tube upwards and traps the rope so one stops.

These devices still require training but the amount of training is reduced and the devices fitted with override make them fool proof in operation. Once more they do use large ropes and the effect of twist coupled with friction reduces rope life. A smaller version using 9mm rope was developed but limited to 25 m.

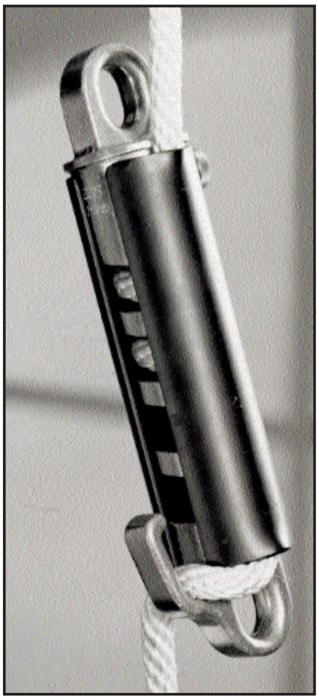


figure A1.15-Spiral device with cylinder, this version is completely manual

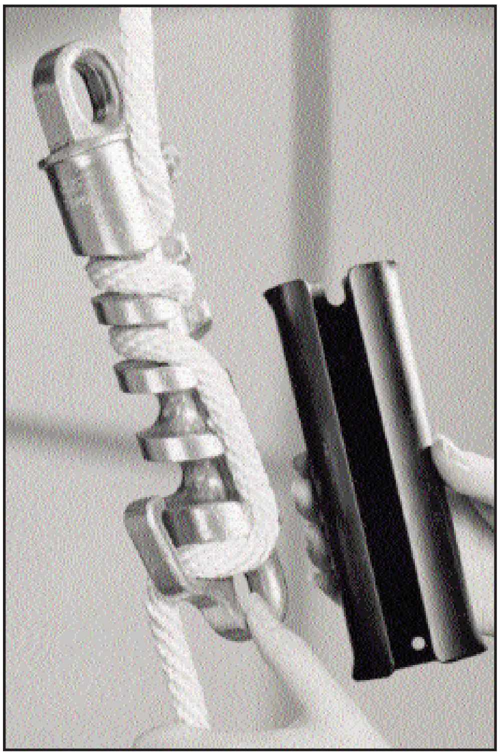


figure A1.16 -Exploded shot showing the centre spindle, it shows two wraps but this can be increased to 3 wraps for slower descents

figure A1.17- Spiral device, however the device is fitted with a stop-go-stop mechanism. In this test the user is in a rescue nappy which can be used for children or injured person. For longer descents it also offers the user a more secure body device than the helicopter strop and is more comfortable than a harness.

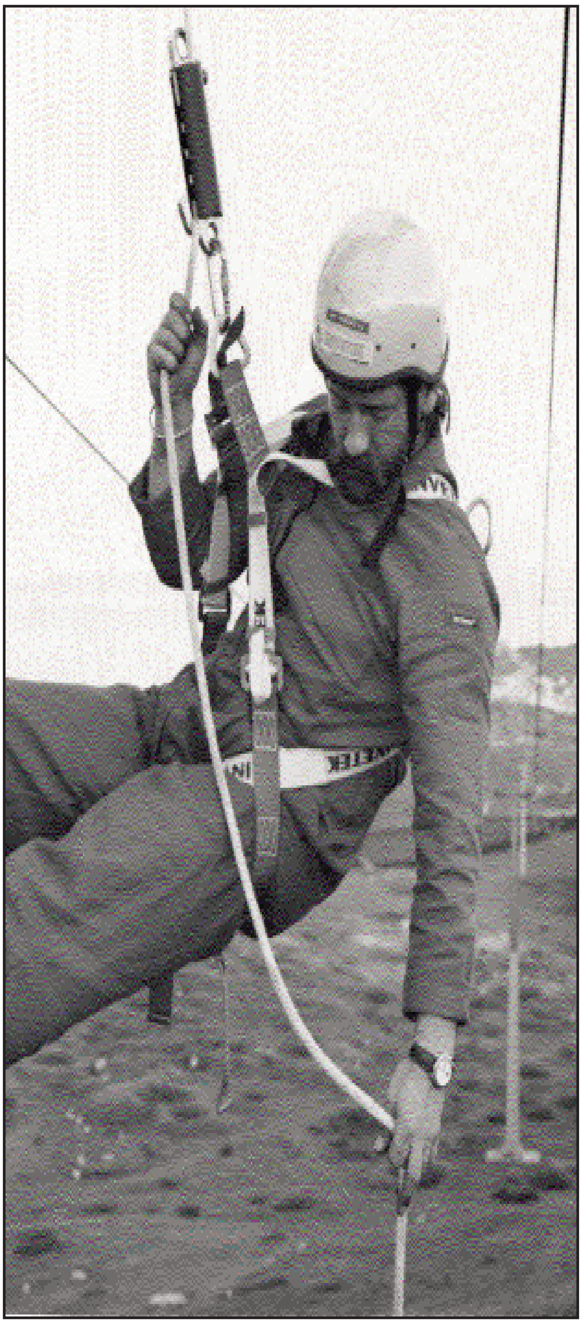


figure A1.18- Spiral device, as per the one shown in figures 15 & 16, easier to use than a figure of 8 however it relies on the user for control, training would be required.

A1.3.7 Cam

In order to address the issues of rope wear and the need for more user control, descent devices are altered to try to stop the excessive spindle wear associated with spiral devices.

The change involved threading the lifeline through a simple device and applying control using a hand lever through the use of cams.

Depending on the threading the user can increase or reduce friction to limit the maximum descent speed. In addition one can add a dead man's brake to the device, if the handle is gripped too hard or released then the descent stops immediately.

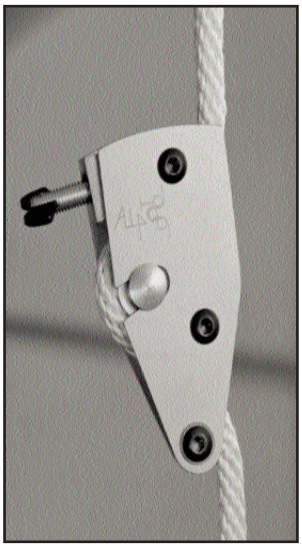


figure A1.19 -type 1 descender, speed is set by the adjusting screw at the top of the device. This type of unit has no dead man's handle and requires training

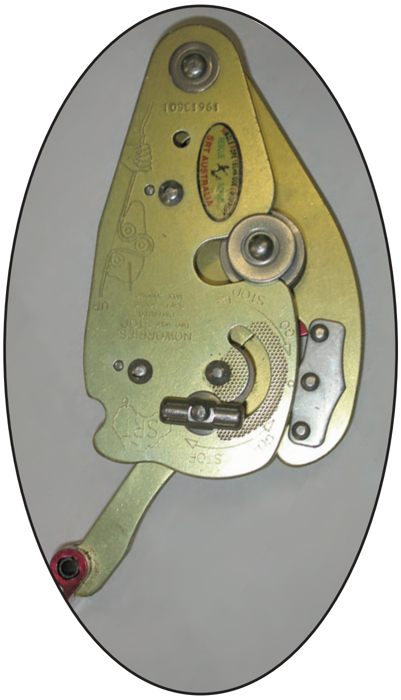


figure A1.20-type 2 descender, controlled by the handle. The device locks if gripped or released.

figure A1.21-type 3 descender, controlled by the handle as with type 2 if you grip the handle or release it the descent is halted.

A1.4 Centrifugal - Reciprocating

In order to gain a greater descent distance the braking mechanisms used in the inertia devices are fitted into a smaller housing by removing the rewind mechanism. With the rewind removed the device can either have a defined spool of lifeline or alternatively the device can have a lifeline that runs through it. As with the inertia device three gear ratios were tried and 12:1 was found to offer the best results. The lower the gear ratio the faster the descent. What was found in this instance was that the unit when developed did not have any of the problems found on the inertia unit when the drop was increased from 15 to 35 m. Being endless, the drum remains engaged eliminating the requirement for a variation of gear ratio. This bodes well for longer descents or units of the same type installed at different levels. The endless lifeline device used a 9mm fiber rope reducing the weight requirement from most of the hand held devices tried. In addition tests have been carried out using a fiber rope with a wire core. By designing the drum ribs in such a manner as to apply a “snake” action to the lifeline, should the fiber rope be burned away or melt the descender will continue to operate in theory.
In the case of the fiber rope, as a user descends the free end rises and this would permit multiple evacuation. Problems occurred with the free end snagging on the return. If there are obstructions this might prove dangerous as the user could get trapped between the anchorage and the ground. In addition as the rope length increased above 50 m the units occasionally went into a stall. The maximum descent is in the region of 100 m depending principally on the user weight and environmental conditions such as rain and wind. If the rope were payed off a reel at the anchorage point then longer descents would be possible but this in turn would limit the device to a maximum of two persons.

figure A1.24- A centrifugal prototype device with no recoil and 9mm lifeline developed by the author

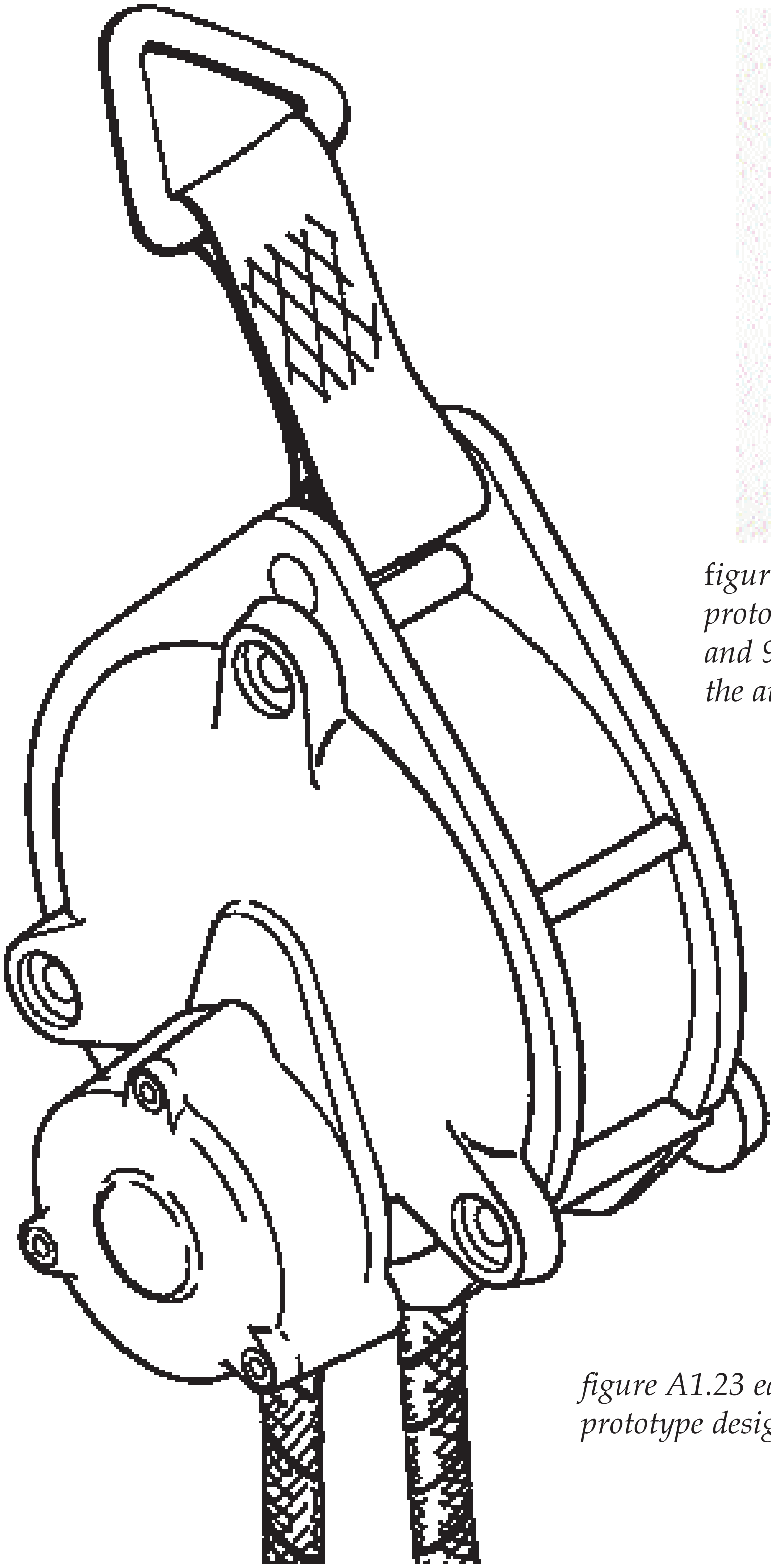


figure A1.23 early prototype design

A1.5 Pulley friction over drum with clutch

The final friction device design to be tried is a pulley system that in general is used for distances of less than 25 m. By using pulleys and a one way clutch hub the ropes are used to raise and lower a user, this can be carried out by the user or by another person who can raise and lower a person from the bottom or top. The clutch hub rotates with the rope in the lift position and generates friction on the lowering cycle to slow the descent. An over-speed device or brake handle is required for lowering as the device will accelerate unless checked.

The design is such that a force of less than 120 N is sufficient to arrest the descent. The smallest rope size used and tested is 9 mm for a single person with 11mm used for two persons. The rope being kernmantle low stretch static rope manufactured from polyester with a tight weave to prevent sheaf slippage. The device has to comply with EN 341-1993 class A.

figure A1.24 - Pulley system under test

figure A1.25- Mark 3 development pulley

A1.6 Frame - friction over frame

The descent is controlled by the action of the webbing wrapping around a frame almost like a gateway, if one holds the free end of the webbing one can control the rate of descent.Tests have been carried out on rack or frame units which are compact and can run with thin tape, tests to date have been carried out to EN341-1993 Class D for 35 m.

This type relies on the user to control the descent. Webbing tested to date is tubular as it offers better wear resistance than single tape as it passes through the device. The type and construction of the tape is a critical factor for the device to work correctly.

figure A1.26- Frame or rack type descender, the webbing strap pulls through the frame, a trial was run using a screw controlled centre barrel to regulate the speed.

A1.7 Discussion relating to main research

The objective is to produce a lightweight device that is robust enough to survive adverse conditions either environmental or artificial whilst being remaining portable. Material choice will play a large part in the final design if a safe and repeatable solution is to be found.

The starting points was to make prototypes using metals as they are expected to be able to work in adverse conditions. After that engineering polymers including glass/mineral reinforced grades and other materials would be tested to see if they could replace or work in harmony with the metal parts.

The cast versions were manufactured by gravity method due principally to cost. However the problem was that section properties had to be thicker than could be obtained using a low or high pressure casting method. As the main influence of this would apply to weight not function it was not considered to be a major issue. Aluminium grade LM25TF in some applications such as off shore installations has restricted use due to its oxidation issues.

Even if anodized it may encounter acceptability problems with a number of areas requiring intrinsically safe devices as any possible spark generation is not permissible. The grade of LM25TF of aluminium was chosen due to its casting properties linked with higher strength and thinner sections when heat treated. Furthermore following heat treatment the machinability of the material is excellent.

The aluminium cast parts lent themselves to be substituted by plastics and a number of materials were tried from Perspex, polycarbonate through various nylon 6 ;11 & 12 grades.

In the end two materials Grilamid and Grivory were selected. In abuse tests they proved very robust and these materials are currently used in adverse conditions such as gearing and chemicals, the latter being used in sheaves. They maintain their mechanical properties at both low and high temperatures. Water absorption is also minimal for the two selected material grades. The Grilamid has another property which may help its acceptance and that is it can be translucent or tinted to assist architectural acceptance and function testing and fault diagnosis.

In phase one the emphasis was to test materials in prototypes to determine descent options and capability up to 35 m. The materials would have to be able to maintain the capability for over 1000 m the current maximum is 60 m using a 4 mm wire rope lifeline which does not comply with EN341 – 1993 as it can not achieve the 12 kN minimum break.
The next phase is to concentrate on the lifeline material itself. At present fiber ropes are 9mm (Kernmantle) and wire is 5 mm diameter (6 x19 and 7 x 19 construction). However the next materials will be 8 mm with wire core, 4mm fiber and 2 and 3 mm diameter wire ropes in order to reduce the device size and weight. The 2 mm wire comes with a breaking strength of 279 kg as a minimum and 550 kg is easily obtained with 3mm.
Considering the standards it may be easier to argue for the 3mm case as it currently exceeds the class D requirement for EN 341- 1993.
Dyneema (5.5mm Diameter) was tested and although it was found to have the strength, shock and other properties the question of fire is a potential issue.

Kevlar and Spectra fiber rope, which are better in heat and flame, suffer from a distinct dislike to bending and shock, which in trying to develop a compact device with maximum length work against its inclusion unless considered in terms of a hybrid mix with other materials.

The webbing material used to date is Polyester due to its good chemical resistance coupled with low stretch. However, a mixed fiber may prove to be better. The thinner the webbing the greater the capacity as demonstrated on the prototype, which brought the inclusion of kevlar.

A1.8 Results and tests - initial

The following is an overview from the tests and trials carried out to date:

A1.8.1 Inertia reel

The devices have been tested to EN 360- 2002 and EN 341- 1993, in the first standard the units have to be tested as a retractable lifeline and in the second they have to be tested to the descender standard.Tests were carried out in accordance with the descender standard class B device.It is a matter of conjecture if the devices have to be tested to the class A level.The nature of the device being such that it is forseen as an emergency unit for one person. In reality it can be used for multiple evacuation greater than 10 persons but that seems unlikely. The units are limited to 60 m currently and all tests were carried out on units of 20 and 35 m and in each case they would require defined anchorages and are not very portable. In the case of the 60 m unit then that is an installed device

.BS EN 360 - 2002.

A1.8.2 Retractable type fall arrestors

The applicable provisions are contained in Clause 4 which deals with the requirements. In the designs made and tested there was a back up fall arrestor that would act as an over speed brake which also had to be tested according to the standard. In the hybrid units testing had to also comply with EN 363 2002 which covers design and ergonomic requirements.

Testing proved compliance with all three standards. The device was then tried over water to see if it performed with different life jackets. The tests were performed in the environmental pool at the Robert Gordon Institute of Technology in Aberdeen. All tests proved the design was acceptable for evacuation into water.

A1.8.3 Figure of 8

The figure of 8 was tried but it required ropes in excess of 10 mm and required a high degree of training in order for it to be used safely. It was tried with a variety of rope and as the figure shows that included the basic three stand nylon. Heat generation during use could be a problem as the user often grips the device either by accident or in a panic situation.

figure A1.27- Figure of 8

A1.8.4 Spiral.

As with the figure of 8 the basic spiral device required skill and tended to wear badly.In fact some older units used by window cleaners on high rise buildings showed wear in excess of 50%, plus the ropes used where badly worn. The improved units with a spring loaded stop and go mechanism eliminated some of the concerns but in use it was found to be too precise.
Volunteers often found them hard to move and in a panic environment this may be a problem.It is also difficult to see how the unit can be made smaller but it is worth including for further trials and testing.

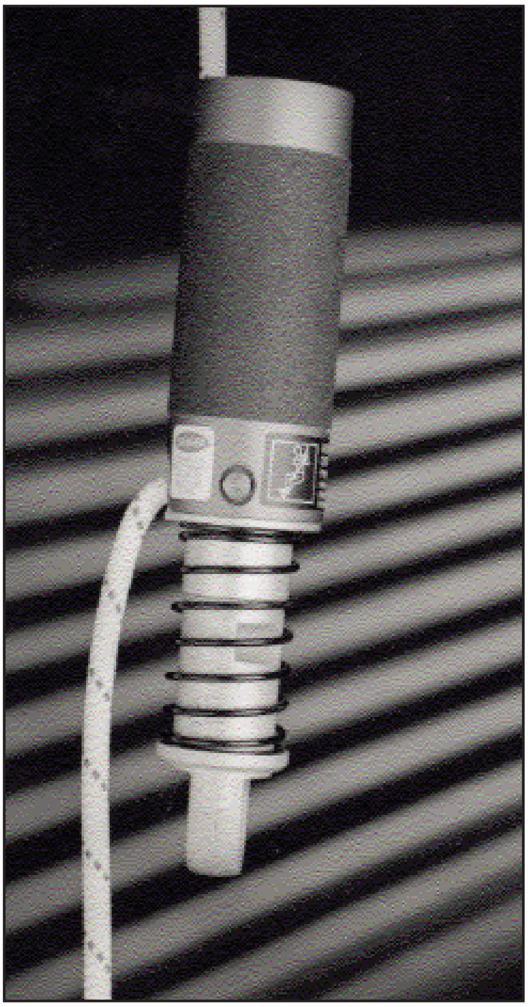


figure A1.28 - shows a spiral device fitted with a stop go stop mechanism. To operate the outer tube is pulled downward to its central position if pulled to hard then it will stop, simply release to stop

A1.8.5 Cam.

A number of cam types of descender were tested. All worked well but all ran on 9 to 13mm kernmantel rope. They all require a degree of training and the ropes would have to investigated as to fibre choice. All were individual descenders. The following are examples of those tested figures A1.29 - A133. None of the devices comply with EN 341 - 1993 as one would have to physically lock the device off to comply with descent speed clause.

figure A1.29- Speed governed by screw at top

figure A1.30

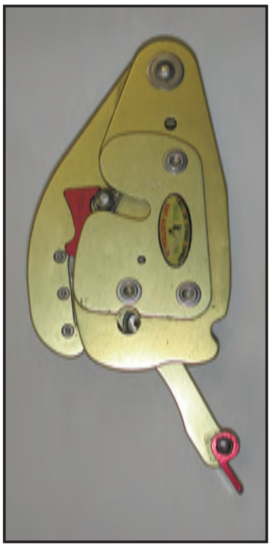


figure A1.31

figure A1.33

figures A1.30 - A1.33 Utilise a handle to act as the speed control, integrated into the handle mechanism is the stop - go - stop or dead man's handle arrangement

A1.8.6 Centrifugal.

A number of types were designed and tested using two main brake methods, the first had two brake shoes pivoted at the bottom and the other models had up to 4 brake shoes that were allowed to slide in each quadrant. The type 1 units failed the tests due to over speeding, despite varying the gear ratio it was not possible to get repeatability. The problem was that when the brake in the inertia unit was made too small it became unstable.

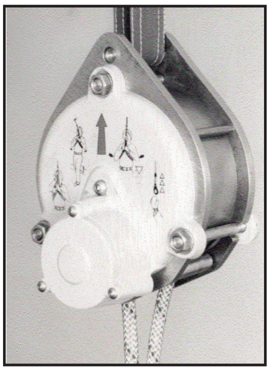


figure A1.34 Type 1 small brake housing using leading and trailing edge brakes

figure A1.35 Type 2 A machined prototype with increased brake housing

figure A1.36 Type 3 A cast solution with finned brake housing

figure A1.37

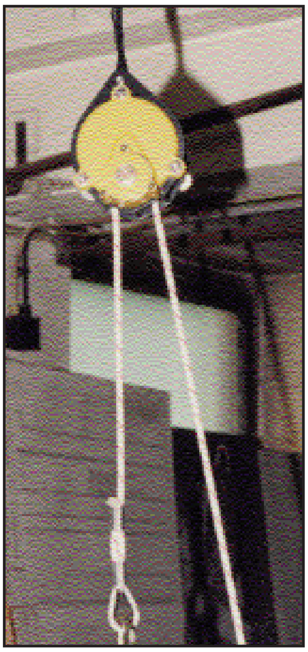


figure A1.38

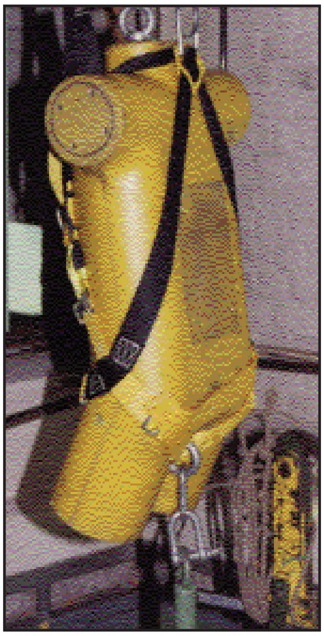


figure A1.39

A1.8.7 Pulley.

Three versions are tested all struggled when descent distances exceeded 30m. Maximum suggested is 50 m. Heat generation and dissipation was a problem with the stainless steel version.

BS EN 341 - 1993

Requirement	Clause	Tested	Result	Comment
Synthetic Fibre Rope Terminations must resist 12kN	4.1.3 4.1.5	Yes Yes	Pass	9 mm Super kernmantel rope with pressed ferrule terminations Terminations slipped at 12.2 kN
Force to hold 80kg load must be < 120 N	4.2	Yes	Pass	A force of 59 N was required
Load to 12 kN for 3 min.	5.5.2	Yes	Pass	Sustained 12 kN for 3 min without damage both before and after descent energy test
To descent energy < 7.5 .E6 J	5.6	Yes	Pass	Sustained the specified descent energy without damage or loss of effectiveness
Temperature rise of device < 48 deg C for parts touched	4.5	Yes	Pass	During the descent energy test the drum of the device reached 83 deg. C. Cheeks 20 C No part can be touched in use
Descent velocity	4.6	Yes	Pass	Descent velocity with 100kg was 1.5m/s
Operation after descent energy test 30 kg load	5.7	Yes	Pass	No change in operation or safety of the descender device was apparent
Operation after descent energy test 150 kg	5.7	Yes	Pass	No change in operation or safety of the descender device was apparent

Table A1.1: Pulley type descender tests to EN 341

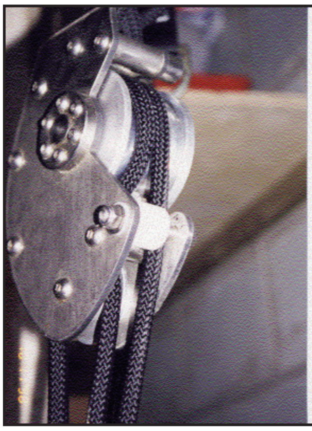


figure A1.40

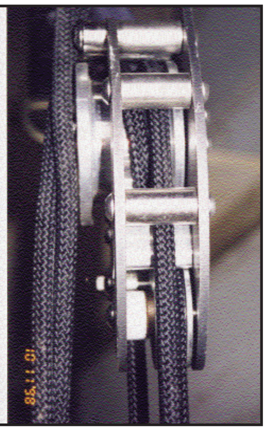


figure A1.41

figure A1.42

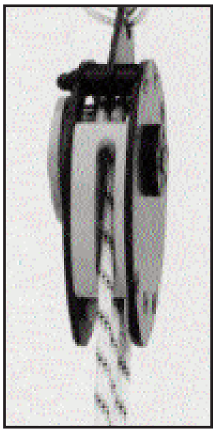


Figure A1.43

figures 1A1.40 and A1.41 : Pulley version 1 heat generation was high due to all stainless steel construction

figures A1.42 and A1.43: Pulley version 2 alloy construction and reduced friction guides

A1.8.8 Frame.

The maximum descent attempted was 35 m. The device did start to warm up. Again it is a device that in use the user should not touch it. Never the less further tests have to be carried out to determine the maximum descent.
The test program has been repeated twice with almost identical results.
The units have been tested with volunteers who found it easy to use. After use the re-threading of the unit was time consuming.

BS EN 341 - 1993

Requirement	Clause	Tested	Result	Comment
Synthetic Fibre Tape Terminations must resist 5kN	4.1.3 4.1.5	Yes Yes	Pass	Synthetic tape with stitched ends, breaking strength rated at 9kN
Force to hold 80kg load must be < 120 N	4.2	Yes	Pass	A very low force of 10 N was required
Load to 5 kN for 3 min.	5.5.2	Yes	Pass	Sustained 5 kN for 3 min without damage both before and after descent energy test
To descent energy < 0.02 .E6 J	5.6	Yes	Pass	Sustained the specified descent energy without damage or loss of effectiveness
Temperature rise of device < 48 deg C	4.5	Yes	Pass	Temperature rise 38 Deg C
Descent velocity	4.6	Yes	Pass	Descent velocity controllable to 2 m/s max max velocity with 100 kg load 1.0 m/s

Table A1.2 : Frame type descender tests to EN

A1.8.9 kernmantle rope flame test

A test is carried out on samples of 9mm rope to determine the time period for the rope to burn through.
The test consisted of tensioning a rope sample and then applying a constant flame from a bunsen burner at a constant distance from the rope. The time was then recorded from the start to the period when the rope had burnt through.
Results:
Test piece 1 - 42 seconds
Test piece 2 - 45 seconds
Test piece 3 - 46 seconds.
By applying a flame retardant chemical to the sheath there was a slight improvement.
Test piece 1 - 49 seconds
Test piece 2 - 50 seconds
Test piece 3 - 49 seconds.
The rope was able to withstand a continuous heat of 150^{O}C and the melting point of the fibre is 250^{O}C.
The chemical impregnation was in accordance with TL 4020 0 023 which is a German government defence department standard. The standard states that the rope should not continue to burn after exposure to the flame is removed.The tests were carried out on two different manufacturers ropes but the results were consistent for both.

A1.9 Discussion and conclusion

From attendance at a number of International trade fairs and symposiums during the last four years including those that have occurred since the September 11th tragedy in both Europe and North America, there has been little evidence of any advances or progress in the sphere of descent control devices. In addition having reviewed the Internet offerings on the same subject matter, once more there appears limited or no action.

The search for published work failed to find any new research. It would appear that in terms of research the major International companies are standing back, concentrating on the products currently available that date back many years.

Moving to the research as it stands today, a number of designs have been developed and evaluated plus a number of other commercial products that are available.

The traditional products which have been around for some time start with the basic figure of 8. This has been a cornerstone device in climbing and abseiling worldwide. Over the years little has changed. A few limbs have been added so that you can stop a descent by tieing off the rope and Titanium has been used for its wear and heat transfer characteristics. a requirement for training, which is considerable, and for the inexperienced there is a high fear factor. Its true to say that the device is well proven but it runs on heavy rope which is a problem and its reliance on training eliminates it from inclusion in any further research.

Climbing companies realised that for many applications the basic figure of 8 was very limiting. What emerged was the need to have more control both in the commercial and military areas. The military wanted to have the capability to abseil but stop quickly in order to enter buildings or use their fire arms. On the commercial front, buildings required window cleaning and not all were fitted with cradles or swing stages from which to work. This led to the development of the spiral and cam devices, which ran on the large ropes but developed to incorporate stop – go – stop mechanisms to reduce training and provide a safer solution coupled with broader application. Again the devices worked well but were still classed as manual. If the user does not operate something then they will not work. The skill factor was not eliminated. In addition the requirement was for bulky ropes which were heavy and hard to carry. Again descents over 50 m would be a problem, the shear bulk of rope making them an unmanageable solution. The device is retained for further work on the basis that it can be scaled down and overcome some of the manual aspects.

Moving to the centrifugal devices, which include the inertia ones, these proved very successful. They are fully automatic taking away the user element completely. They do have the problem of descending into a danger and that requires further analysis and application.

The inertia devices have the additional problem of the rewind mechanism, which is heavy and bulky limiting the device to 60 m. There is only one 60 m unit available and the design of that is unrefined. It works in a fashion and the application it is used for is angled evacuation form the monkey board of an offshore drilling installation (a monkey board is approximately 33 m up the drilling tower where a person has to walk out along a plank to guide the drill tube sections into the drill head). The other units are for 35 m or less and they have multiple uses but would require a major re think on the rewind mechanism to achieve or come close to the 100 m plus objectives of this research. Taking the braking mechanisms out of the inertia devices does have potential as they can be used with either integral lifelines or a continuous spool lifeline. This would extend the length considerably. The heat build up was not excessive in the devices tested and there was no initial evidence of brake fade which is promising. If one compares this to the hydraulic devices they also performed well and have similar feature to the centrifugal type. There are questions regarding the sealing and effect of temperature on the unit performance and the oil which require answers but initial tests are good and the designs can be looked at and developed to a final solution. Considering the lifeline aspects, if one takes the drum out of a 35 m inertia device and change to the possible 2mm lifeline that is being considered then the descent distance increases from 35 m to 500 m. With a redesign of the device this distance could be increased further.

The pulley designs tested have limited application, as they require a considerable amount of rope and struggle above 25 m with a maximum use of 50 m. The designs tested were running on 9 mm diameter rope. They work well in rescue and have the ability to ascend as well as descend but for high-level evacuation they are omitted from further work.

The final device tested was the frame, which performed well and is automatic but with a manual override. The design runs on webbing, which can be compacted into a spool arrangement and there could be a supply of spools that can be easily fitted.

Appendix 2 - Results

Included in this section

A2.1 SATRA technical services report - SPC0166172/0830/2/NW Issue 2
Subject - testing of descender refernce 1270 in accordance with EN 341

A2.2 SATRA technical services report - SPC0165653/0828/NW
Subject - testing of descender devices in accordance with EN 341

A2.3 SATRA technical services report - SPC0166172/0830/1/NW Issue 2
Subject - testing of a descender reference 1210 in accordance with EN 341

Appendix 2 - Results - continued

Tables of test data included in this section

Figures included in this section

A2.1 SATRA technical services report- SPC0165653/0828/NW

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Firm: Descent Control Ltd
Knowsley Business Park
Knowsley
Merseyside
L34 9ET

For the attention of: Mr M Worsley

Technical Services Report

Subject: TESTING OF DESCENDER REFERENCE "1270" IN ACCORDANCE WITH EN 341: 1992
Firm: Descent Control Ltd
Our ref: SPC0166172/0830/2/NW Issue 2
Your ref:
Date: 8 September 2008

Conditions of Issue:

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Except where stated, an uncertainty has been applied to the results within this report, based on a standard uncertainty multiplied by a coverage factor k = 2, providing for a confidence level of approximately 95%

Tests marked † are not UKAS accredited.

(Page 1 of 6)

SATRA Technology Centre Ltd. A subsidiary of SATRA. Registered in England No. 3856296

Technical Services Report

INTRODUCTION

Samples of descender, reference "1270", were received by SATRA on 25 July & 15 August 2008, for testing in accordance with EN 341: 1992 as a class C device. Testing was carried out on 25 July & 5 September 2008, in the presence of Mr Stephen Griffiths, of AYD Ltd, and on 28 August 2008.

CONCLUSIONS

The samples of descender, reference "1270", as received by SATRA on 25 July & 15 August 2008, have been tested to EN 341: 1992 as a class C device, and found to achieve the following requirements:

SAMPLE REFERENCE	STANDARD	CLAUSE / TEST	PASS / FAIL
1270	EN 341: 1992	4.1.1 Ropes, straps	PASS
		4.1.2 Wire ropes	N/A
		4.1.3 Synthetic fibre ropes	See note 2
		4.1.4 Straps	N/A
		4.1.5 Terminations	See note 2
		4.2 Holding load of hand operated descender devices	N/A
		4.3 Static strength	PASS See note 3
		4.4 Descent energy	PASS See note 1
		4.5 Temperature rise of the descender device	PASS See note 1
		4.6 Descent velocity	PASS See note 1
		4.7 Special requirements for devices class D	N/A

Note 1 – Testing carried out & originally reported under SATRA reference SPC0165653/0828/NW

Note 2 – Clauses marked with 'not assessed' must be addressed before an EC type examination certificate can be issued

Note 3 – Testing carried out under SATRA job reference SPC0167327/0836

Signed: N Wright
PPE Technologist
Safety Product Centre

Technical Services Report

TEST RESULTS

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.1.1 Ropes, straps	Ropes shall consist of synthetic fibres or steel wires	Synthetic rope used	PASS
4.1.2 Wire ropes	Wire ropes shall be made from galvanised steel wire Wire ropes shall be stress and torsion relieved Wire ropes shall be made from one piece The nominal tensile strength of the rope should be 1770N/mm^2, and shall not exceed 1960N/mm^2 Ropes shall be of a type capable of visual inspection or else subject	Not applicable	N/A
4.1.3 Synthetic fibre ropes	Ropes shall be designed in a core-case plait Ropes shall be made from polyamide or a material of the same quality The displacement of the case shall not exceed 15 mm over a rope length of 2 m The elongation shall not exceed 8 % Ropes of different design may be used for descender devices class D	Not assessed Not assessed Not assessed Not assessed Not applicable	See note 2
4.1.4 Straps	Straps shall be made of a material equal in quality to ropes made from synthetic fibres	Not applicable	N/A

Technical Services Report

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.1.5 Terminations	Terminations shall exclusively be made by the manufacturer or a person authorized by the manufacturer	Ropes terminated with encased loops and attached connectors	PASS
	Terminations shall be designed in a way that they can only be opened by means of a tool	Terminations cannot be opened without damaging ropes	PASS
	Terminations shall be marked by the manufacturer	Not assessed	See note 2
	Terminations of synthetic fibre ropes shall be made with knots or with pressed ferrules	Terminations formed with encased knots	PASS
	The rope including terminations shall resist a static strength of 12 kN (Class D 5 kN) for a period of 3 min.	Not assessed due to failure of casing	See note 2
4.2 Holding load of hand operated descender devices	The maximum load necessary for holding the mass fixed at the end of the rope leaving the device shall be 120 N	Not applicable	N/A
4.3 Static strength	After a load of 12 kN (Class D: 5 kN) has been applied, no part of the device shall show any signs of breaking or tearing	'Quenched' casing version: 12 kN sustained for 3 minutes without failure Force increased to 14.2 kN before fracture of handle (See figure 1)	PASS See note 3
		'Dry' casing version: 12 kN sustained for 3 minutes without failure Force increased to 15.9 kN before fracture of handle (See figure 2)	

Technical Services Report

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.4 Descent energy	Descender device shall resist the descent energy determined for its class without any impairment of safety Class A: $W \geq 7.5 \times 10^6$ J Class B: $W \geq 1.5 \times 10^6$ J Class C: $W \geq 0.5 \times 10^6$ J Class D: $W \geq 0.02 \times 10^6$ J	Test mass: 75 kg Descent height: 43 m Number of descents: 16 Descender device found to resist a descent energy of approximately 506,196 J without any impairment of safety No visual evidence of changes to webbing or device following descent energy test	PASS See note 1
	After test for descent energy and functional test, the device and rope or strap shall not show any changes affecting its safety	Descender device found to allow 30 kg mass to descend under full control Descender device found to allow 150 kg mass to descend under full control	PASS
4.5 Temperature rise of the descender device	The temperature due to friction shall not rise to a point affecting the function of the device None of the parts of the device touched during the descent shall develop a temperature higher than 48 °C	Device was found to pass function and descent energy tests, thus temperature rise not considered to affect device function Peak temperature of device during descent energy test: 42.8°C	PASS See note 1
4.6 Descent velocity	It shall be possible to keep the descent velocity between 0.5 m/s and 2 m/s (Class A, B, C) For devices of class D it shall be possible to maintain a descent velocity of 2 m/s In the case of hand operated devices the velocity must not exceed 2 m/s after the control device was released During the descent the descent velocity shall be almost constant	Minimum speed of descent: 0.65 m/s Maximum speed of descent: 0.83 m/s Not applicable Not applicable Descent velocity maintained at near-constant rate	PASS See note 1
4.7 Special requirements for devices class D	Descender devices of class D shall be designed in such a way that they cannot be used more than once	Not applicable	N/A

A2.2 SATRA technical services report- SPC016172/0830/2/NW Issue 2

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Firm: AYD Ltd
Hendy
Waunfawr
Near Caernarfon
Gwynedd
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For the attention of: Mr S Griffiths

Technical Services Report

Subject: TESTING OF DESCENDER DEVICES IN ACCORDANCE WITH EN 341: 1992
Firm: AYD Ltd
Our ref: SPC0165653/0828/NW
Your ref:
Date: 18 July 2008

Conditions of Issue:

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Except where stated, an uncertainty has been applied to the results within this report, based on a standard uncertainty multiplied by a coverage factor k = 2, providing for a confidence level of approximately 95%

Tests marked † are not UKAS accredited.

(Page 1 of 5)

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SATRA TECHNOLOGY CENTRE

Technical Services Report

INTRODUCTION

Samples of Descenders, reference "1210" & "1270", were received by SATRA on 15 July 2008, for testing in accordance with EN 341: 1992 as class D and C devices respectively. Testing was carried out on 15 July 2008, at Liverpool Bulk (EON Coal Terminal). All testing was overseen by Nathan Wright of SATRA Technology Centre Ltd, with all measurements taken or verified using SATRA calibrated equipment.

CONCLUSIONS

The samples of Descenders, reference "1210" & "1270", as received by SATRA on 15 July 2008, have been tested to EN 341: 1992, and found to achieve the following requirements:

SAMPLE REFERENCE	STANDARD	CLAUSE / TEST	PASS / FAIL
1210	EN 341: 1992	4.4 Descent energy	Not fully assessed
		4.5 Temperature rise of the descender device	PASS
		4.6 Descent velocity	Not fully assessed
1270		4.4 Descent energy	**FAIL**
		4.5 Temperature rise of the descender device	Not fully assessed
		4.6 Descent velocity	PASS

Note 1 – Clauses marked as 'not assessed' must be addressed before an EC type examination certificate can be issued

AYD Ltd
SPC0165653/0828/NW
18 July 2008

(Page 2 of 5)

Signed: N Wright
PPE Technologist
Safety Product Centre

SATRA TECHNOLOGY CENTRE

Technical Services Report

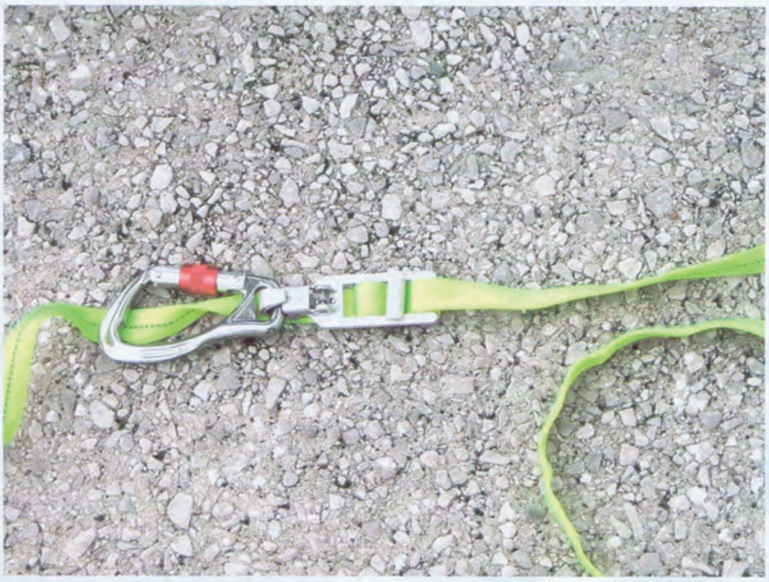

Figure 1 – Descender device reference "1210"

Figure 2 – Descender device reference "1270"

AYD Ltd
SPC0165653/0828/NW
18 July 2008

(Page 3 of 5)

Signed: N Wright
PPE Technologist
Safety Product Centre

Technical Services Report

TEST RESULTS

Table 1 – Testing of descender reference "1210" in accordance with EN 341: 1992

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.4 Descent energy	Descender device shall resist the descent energy determined for its class without any impairment of safety Class A: $W \geq 7.5 \times 10^6$ J Class B: $W \geq 1.5 \times 10^6$ J Class C: $W \geq 0.5 \times 10^6$ J Class D: $W \geq 0.02 \times 10^6$ J After test for descent energy and functional test, the device and rope or strap shall not show any changes affecting its safety	Test mass: 100 kg Descent height: 43 m (Descender intended for 35 m descent height) Descender device found to resist a descent energy of approximately 42,183 J without any impairment of safety No visual evidence of changes to webbing or device following descent energy test Note – function test using 30 kg test mass not carried out	Not fully assessed
4.5 Temperature rise of the descender device	The temperature due to friction shall not rise to a point affecting the function of the device None of the parts of the device touched during the descent shall develop a temperature higher than 48 °C	Device function not affected at any point during descent User does not touch descender device during descent	PASS
4.6 Descent velocity	It shall be possible to keep the descent velocity between 0.5 m/s and 2 m/s (Class A, B, C) For devices of class D it shall be possible to maintain a maximum descent velocity of 2 m/s In the case of hand operated devices the velocity must not exceed 2 m/s after the control device was released During the descent the descent velocity shall be almost constant	Not applicable Descent velocity maintained at between 0.5 m/s & 1.5 m/s during testing Not assessed Descent velocity dependant on rate of feed of webbing (controlled by user) – velocity can be maintained at a constant rate	Not fully assessed

SATRA TECHNOLOGY CENTRE

Technical Services Report

Table 2 – Testing of descender reference "1270" in accordance with EN 341: 1992

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.4 Descent energy	Descender device shall resist the descent energy determined for its class without any impairment of safety Class A: $W \geq 7.5 \times 10^6$ J Class B: $W \geq 1.5 \times 10^6$ J Class C: $W \geq 0.5 \times 10^6$ J Class D: $W \geq 0.02 \times 10^6$ J After test for descent energy and functional test, the device and rope or strap shall not show any changes affecting its safety	Test mass: 75 kg Descent height: 43 m Number of descents: 16 Descender device found to resist a descent energy of approximately 506,196 J without any impairment of safety No visual evidence of changes to webbing or device following descent energy test **Device brake found to fail under 150 kg function test – rope pulled through at high speed** Note – function test using 30 kg test mass not carried out	**FAIL**
4.5 Temperature rise of the descender device	The temperature due to friction shall not rise to a point affecting the function of the device None of the parts of the device touched during the descent shall develop a temperature higher than 48 °C	Not assessed due to failure in function test Peak temperature of device during descent energy test: 42.8°C	Not fully assessed
4.6 Descent velocity	It shall be possible to keep the descent velocity between 0.5 m/s and 2 m/s (Class A, B, C) For devices of class D it shall be possible to maintain a descent velocity of 2 m/s In the case of hand operated devices the velocity must not exceed 2 m/s after the control device was released During the descent the descent velocity shall be almost constant	Minimum speed of descent: 0.65 m/s Maximum speed of descent: 0.83 m/s Not applicable Not applicable Descent velocity maintained at near-constant rate	PASS

**************** END OF REPORT ****************

AYD Ltd
SPC0165653/0828/NW
18 July 2008

(Page 5 of 5)

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For the attention of: Mr M Worsley

Technical Services Report

Subject: TESTING OF DESCENDER REFERENCE "1210" IN ACCORDANCE WITH EN 341: 1992
Firm: Descent Control Ltd
Our ref: SPC0166172/0830/1/NW Issue 2
Your ref:
Date: 8 September 2008

Conditions of Issue:

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Tests marked † are not UKAS accredited.

(Page 1 of 5)

Technical Services Report

INTRODUCTION

Samples of descender, reference "1210", were received by SATRA on 25 July 2008, for testing in accordance with EN 341: 1992 as a class D device. Testing was carried out on 25 July & 5 September 2008, in the presence of Mr Stephen Griffiths, of AYD Ltd, and on 28 August 2008.

CONCLUSIONS

The samples of descender, reference "1210", as received by SATRA on 25 July 2008, have been tested to EN 341: 1992 as a class D device, and found to achieve the following requirements:

SAMPLE REFERENCE	STANDARD	CLAUSE / TEST	PASS / FAIL
1210	EN 341: 1992	4.1.1 Ropes, straps	PASS
		4.1.2 Wire ropes	N/A
		4.1.3 Synthetic fibre ropes	N/A
		4.1.4 Straps	PASS
		4.1.5 Terminations	See note 2
		4.2 Holding load of hand operated descender devices	PASS
		4.3 Static strength	PASS
		4.4 Descent energy	PASS See note 1
		4.5 Temperature rise of the descender device	PASS See note 1
		4.6 Descent velocity	PASS See note 3
		4.7 Special requirements for devices class D	PASS See note 3

Note 1 – Testing carried out & originally reported under SATRA reference SPC0165653/0828/NW

Note 2 – Clauses marked with 'not assessed' must be addressed before an EC type examination certificate can be issued

Note 3 – Testing carried out under SATRA job reference SPC0167327/0836

Signed: N Wright
PPE Technologist
Safety Product Centre

N Wright

Technical Services Report

TEST RESULTS

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.1.1 Ropes, straps	Ropes shall consist of synthetic fibres or steel wires	Synthetic webbing used	PASS
4.1.2 Wire ropes	Wire ropes shall be made from galvanised steel wire Wire ropes shall be stress and torsion relieved Wire ropes shall be made from one piece The nominal tensile strength of the rope should be $1770 N/mm^2$, and shall not exceed $1960 N/mm^2$ Ropes shall be of a type capable of visual inspection or else subject	Not applicable	N/A
4.1.3 Synthetic fibre ropes	Ropes shall be designed in a core-case plait Ropes shall be made from polyamide or a material of the same quality The displacement of the case shall not exceed 15 mm over a rope length of 2 m The elongation shall not exceed 8 % Ropes of different design may be used for descender devices class D	Not applicable	N/A
4.1.4 Straps	Straps shall be made of a material equal in quality to ropes made from synthetic fibres	Webbing was found to PASS static strength requirements of EN 341: 1992 Clause 4.3, thus considered to be of adequate quality	PASS

Signed: N Wright
PPE Technologist
Safety Product Centre

SATRA TECHNOLOGY CENTRE

Technical Services Report

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.1.5 Terminations	Terminations shall exclusively be made by the manufacturer or a person authorized by the manufacturer	Ends of webbing terminated with sewn loops	PASS
	Terminations shall be designed in a way that they can only be opened by means of a tool	Terminations cannot be opened without damaging webbing / stitching	PASS
	Terminations shall be marked by the manufacturer	Not assessed	See note 2
	Terminations of synthetic fibre ropes shall be made with knots or with pressed ferrules	Not applicable	N/A
	The rope including terminations shall resist a static strength of 12 kN (Class D 5 kN) for a period of 3 min.	Webbing (including terminations) was found to resist 5 kN for a period of 3 minutes (tested with device)	PASS
4.2 Holding load of hand operated descender devices	The maximum load necessary for holding the mass fixed at the end of the rope leaving the device shall be 120 N	Peak holding load for 100 kg test mass = 80 N	PASS
4.3 Static strength	After a load of 12 kN (Class D: 5 kN) has been applied, no part of the device shall show any signs of breaking or tearing	5 kN sustained for 3 minutes without failure	PASS
4.4 Descent energy	Descender device shall resist the descent energy determined for its class without any impairment of safety Class A: $W \geq 7.5 \times 10^6$ J Class B: $W \geq 1.5 \times 10^6$ J Class C: $W \geq 0.5 \times 10^6$ J Class D: $W \geq 0.02 \times 10^6$ J After test for descent energy and functional test, the device and rope or strap shall not show any changes affecting its safety	Test mass: 100 kg Descent height: 43 m (Descender intended for 35 m descent height) Descender device found to resist a descent energy of approximately 42,183 J without any impairment of safety No visual evidence of changes to webbing or device following descent energy test	PASS See note 1
		Descender device found to allow 30 kg mass to descend under full control Descender device found to allow 150 kg mass to descend under full control	PASS

Descent Control Ltd
SPC0166172/0830/1/NW Issue 2
8 September 2008

Signed: N Wright
PPE Technologist
Safety Product Centre

Technical Services Report

EN 341: 1992 CLAUSE / TEST	EN 341: 1992 REQUIREMENT	RESULT / COMMENT	PASS / FAIL
4.5 Temperature rise of the descender device	The temperature due to friction shall not rise to a point affecting the function of the device None of the parts of the device touched during the descent shall develop a temperature higher than 48 °C	Device function not affected at any point during descent User does not touch descender device during descent	PASS See note 1
4.6 Descent velocity	It shall be possible to keep the descent velocity between 0.5 m/s and 2 m/s (Class A, B, C) For devices of class D it shall be possible to maintain a descent velocity of 2 m/s In the case of hand operated devices the velocity must not exceed 2 m/s after the control device was released During the descent the descent velocity shall be almost constant	Not applicable Descent velocity maintained at between 0.5 m/s & 1.5 m/s during testing (See note 1) Speed maintained below 2 m/s when trailing webbing is released. Test mass eventually halted (See note 3) Descent velocity dependant on rate of feed of webbing (controlled by user) – velocity can be maintained at a constant rate	PASS
4.7 Special requirements for devices class D	Descender devices of class D shall be designed in such a way that they cannot be used more than once	Following one descent, webbing would need to be fed back through device and back into bag, thus making reuse immediately obvious	PASS See note 3

* * * * * * * * * * * * * * * * END OF REPORT * * * * * * * * * * * * * * * *

Descent Control Ltd
SPC0166172/0830/1/NW Issue 2
8 September 2008

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Signed: N Wright
PPE Technologist
Safety Product Centre

A2.4 Test results and graphs

Speed Rev/s	Torque 4S	Torque A1 4L	Torque A1 2L+2T	Torque A1 4T	Torque A2 4L	Torque A2 2L+2T	Torque A2 4T	Power 4S
0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
5	0.3	0.4	0.3	0.2	0.6	0.4	0.2	0.0
10	1.2	1.6	1.1	0.6	2.4	1.5	0.6	0.1
15	2.8	3.7	2.5	1.4	5.4	3.4	1.4	0.3
20	4.9	6.6	4.5	2.5	9.6	6.0	2.5	0.6
25	7.7	10.2	7.1	3.9	15.0	9.5	4.0	1.2
30	11.1	14.7	10.2	5.7	21.5	13.6	5.7	2.1

Table A2.1- Brake speed v power 4 brakes

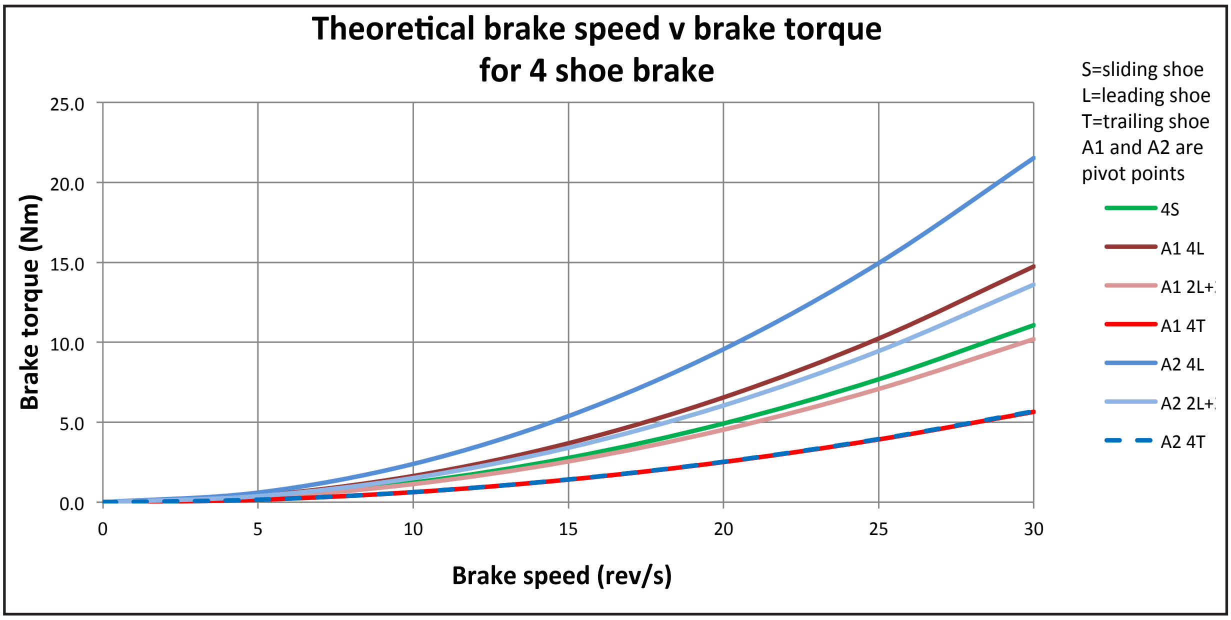


figure A2.1 - Brake speed v power 4 brakes

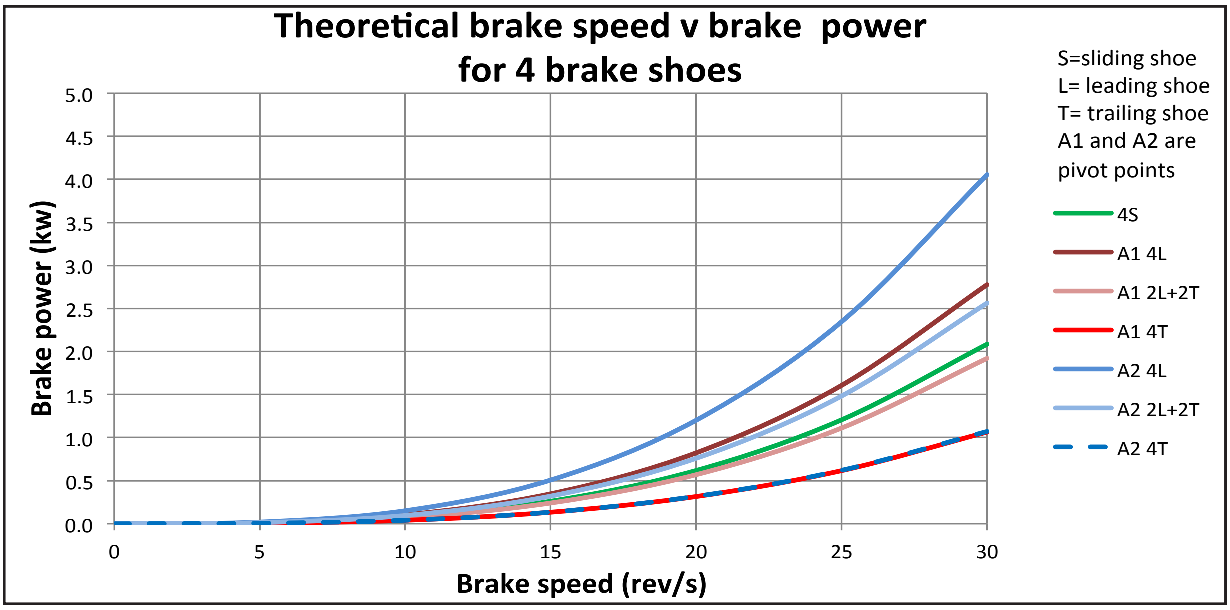


figure A2.2- Brake speed v power 4 brakes

Dynamic tests 75 kg Liverpool 200 1270			
test	time		
	75kg		75kg
1	54	1	0.81
2	55	2	0.8
3	56	3	0.79
4	55	4	0.8
5	56	5	0.79
6	58	6	0.76
7	58	7	0.76
8	62	8	0.71
9	62	9	0.71
10	63	10	0.7
11	52	11	0.85
12	62	12	0.71
13	58	13	0.76
14	66	14	0.67
15	55	15	0.8
16	61	16	0.72

Table A2.2-Dynamic drop results Liverpool

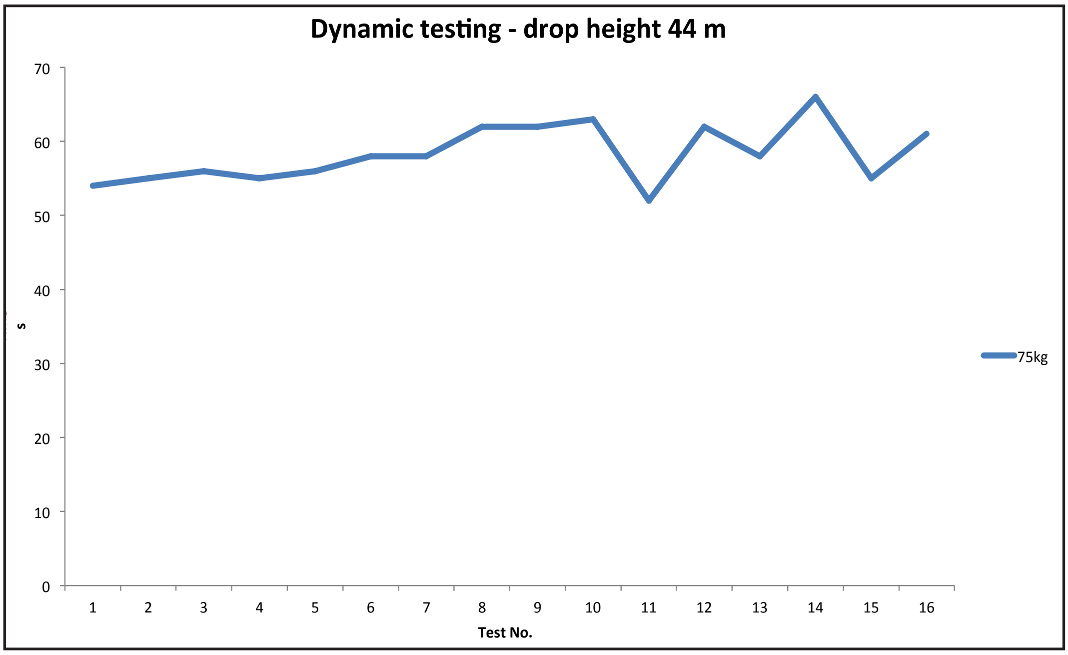


figure A2.3- Liverpool drop - test v time

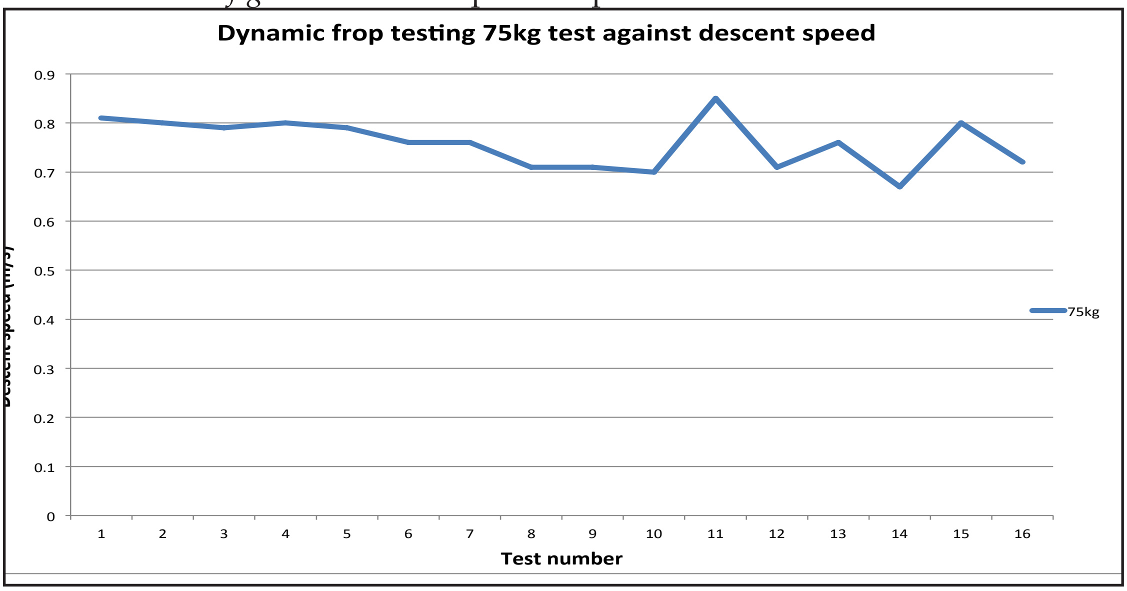


figure A2.4 - Liverpool drop - test v descent speed

Time	ke speed (re	ke torque (N	ke speed (re	ke torque (N	ke speed (re	ke torque (N	Theo2T	TheoL+T	Theo2L	mu=	0.32
ms		2T		L+T		2L				a=	0.023
										b=	0.016
45	0.1	0	0.1	0	0.1	0	0.00	0.00	0.00		
401	0.1	0	0.1	0	0.1	0	0.00	0.00	0.00		
908	0.1	0	0.1	0	0.1	0	0.00	0.00	0.00		
1220	0.1	0	0.1	0	1.4	0.1	0.00	0.00	0.01		
1667	1.6	0.2	0.1	0	2.3	0.6	0.00	0.00	0.02		
2003	3.1	0.4	0.6	0.3	3.6	0.5	0.01	0.00	0.05		
2402	4.6	0.4	3.3	0.6	4.8	0.5	0.03	0.03	0.09		
2806	5.9	0.3	4.6	0.4	7.4	0.6	0.05	0.06	0.21		
3243	8.3	0.2	5.9	0.4	8.7	0.7	0.10	0.09	0.29		
3603	9.5	0.4	7.5	0.7	9.9	0.7	0.13	0.15	0.38		
4059	11	0.5	9.9	0.8	12.7	1	0.17	0.26	0.62		
4402	12.3	0.5	11.3	0.9	13.8	1.1	0.22	0.34	0.73		
4819	14.9	0.6	12.6	1	15.1	1.3	0.32	0.42	0.88		
5307	16.2	0.7	15.1	1	16.4	1.3	0.37	0.60	1.04		
5619	17.4	0.6	16.5	1.2	18.9	1.7	0.43	0.72	1.38		
6067	19.9	0.7	17.7	1.4	20.1	1.7	0.56	0.83	1.56		
6403	21.2	0.9	19	1.5	21.3	2	0.64	0.95	1.75		
6867	22.5	1	21.3	1.5	23.9	2.2	0.72	1.20	2.20		
7202	23.7	0.9	22.5	1.6	25.4	2.2	0.80	1.34	2.49		
7600	25	1.2	24.1	1.9	26.9	2.8	0.89	1.53	2.79		
8001	27.6	1.2	25.4	1.9	28.1	3	1.09	1.70	3.04		
8489	29	1.5	27.8	2.1	29.2	3.5	1.20	2.04	3.28		
8801	30.2	1.4	29.1	2.2	30.5	3.8	1.30	2.23	3.58		
9249	33	1.7	30.3	2.5	32.1	4.1	1.55	2.42	3.97		
9602	34	1.7	31.4	2.5	33.5	4.2	1.65	2.60	4.32		
10049	35.3	1.9	33.9	2.8	36	4.7	1.78	3.03	4.99		
10400	36.3	2	35.2	3.1	37.5	4.8	1.88	3.27	5.42		
10809	38.8	2.2	36.3	3.3	38.4	5.5	2.14	3.48	5.68		
11297	40	2.4	38.8	3.6	39.6	5.8	2.28	3.97	6.04		
11609	40.6	2.5	39.7	4.1	39.9	5.8	2.35	4.16	6.13		

Table A2.3 - 2 shoes - speed v torque data

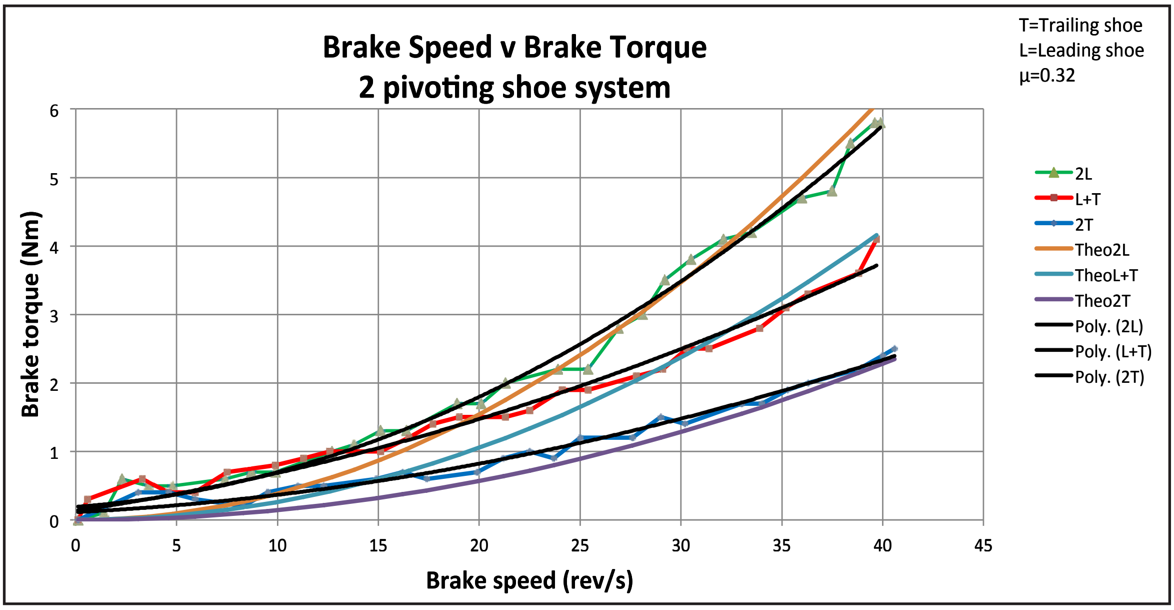


figure A2.5- 2 shoes - speed v torque

rake spee	Brake torqu	Brake spee	Brake torqu	Brake spe	Brake tor		Brake tor		Theo tor	Theo tor
2	A2	A1	A1	A2	A2-5		A1-6		A2	A1
1.8	5.9	3.5	6.7	1.8	0.9		0.7		0.1	0.2
3.2	6.1	5.1	7.2	3.2	1.1		1.2		0.2	0.4
5	6.7	6.3	7.2	5	1.7		1.2		0.6	0.7
7.5	7	7.9	7.8	7.5	2		1.8		1.3	1.0
8.7	7.6	9.3	7.8	8.7	2.6		1.8		1.8	1.4
10	8.1	11.2	8.3	10	3.1		2.3		2.4	2.1
11.3	8.9	12.7	9.4	11.3	3.9		3.4		3.1	2.6
14.1	9.2	14	9.2	14.1	4.2		3.2		4.8	3.2
15.6	9.8	16.5	12.2	15.6	4.8		6.2		5.8	4.5
16.7	10.8	17.7	10.8	16.7	5.8		4.8		6.7	5.1
18.6	10.8	17.7	11.5	18.6	5.8		5.5		8.3	5.1
19.9	12.9	21.1	12.3	19.9	7.9		6.3		9.5	7.3
21.2	14.3	22.7	14.4	21.2	9.3		8.4		10.7	8.4
22.5	15.3	23.7	14.7	22.5	10.3		8.7		12.1	9.2
24.6	16.2	24.8	16.6	24.6	11.2		10.6		14.5	10.1
25.7	17.7	26	17.2	25.7	12.7		11.2		15.8	11.1
26.6	18.6	26.8	18.3	26.6	13.6		12.3		16.9	11.8
27.6	19.4	27.8	18.4	27.6	14.4		12.4		18.2	12.7
28.2	19.3	28.2	18.5	28.2	14.3		12.5		19.0	13.0
28.2	19.1	28.1	18.1	28.2	14.1		12.1		19.0	12.9
28.7	19.1	27.7	17.3	28.7	14.1		11.3		19.7	12.6
		28.1	17.4				11.4			12.9
		28.6	17.8				11.8			13.4
		28.4	18				12			13.2
		29.2	18.4				12.4			

Table A2.4 -4 leading shoes pivot A1 & A2 speed v torque

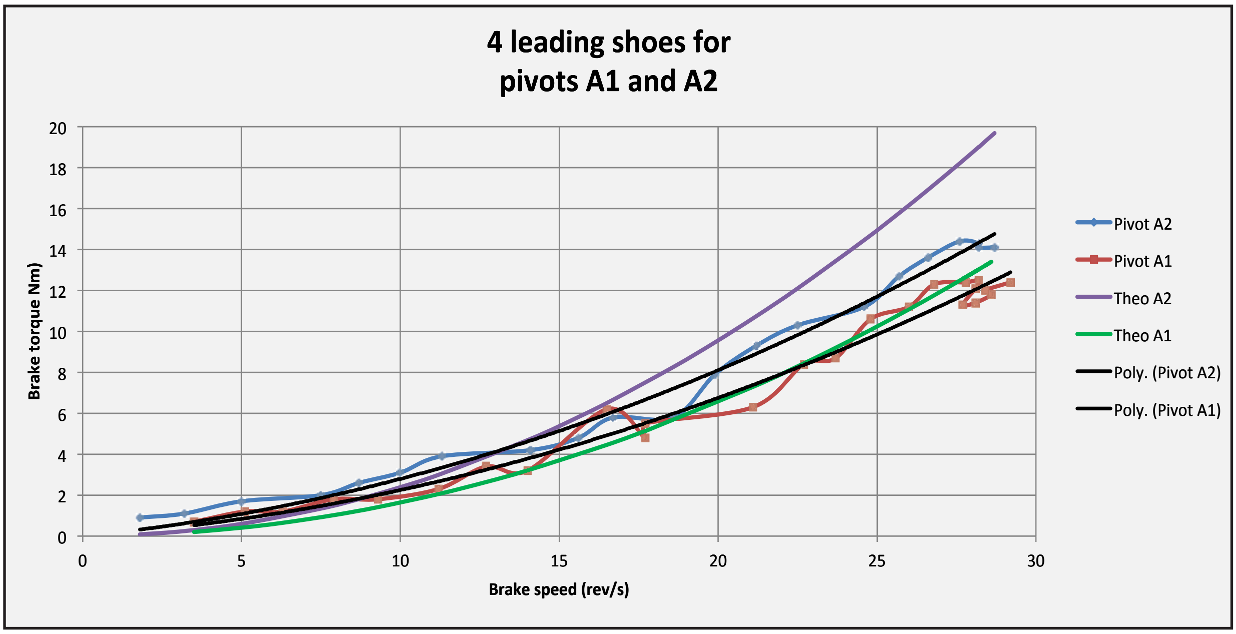


figure A2.6- 4 leading shoes pivot A1 & A2 speed v torque

Length	Velocity	Time	Velocity	Time	Velocity	Time	Length
	Mass 150	Mass 150	Mass 50	Mass 50	Mass 100	Mass 100	
0	2.48	0.00	1.43	0.00	2.02	0.00	0
3	2.40	1.23	1.39	2.13	1.96	1.51	3
6	2.33	2.50	1.35	4.32	1.90	3.06	6
9	2.25	3.81	1.30	6.59	1.84	4.66	9
12	2.18	5.16	1.26	8.93	1.78	6.31	12
15	2.10	6.56	1.21	11.36	1.72	8.03	15
18	2.02	8.02	1.17	13.88	1.65	9.81	18
21	1.93	9.54	1.12	16.50	1.58	11.67	21
24	1.85	11.13	1.07	19.24	1.52	13.60	24
27	1.76	12.80	1.02	22.10	1.45	15.63	27
30	1.67	14.55	0.97	25.11	1.37	17.76	30
33	1.58	16.39	0.92	28.29	1.30	20.00	33
36	1.49	18.35	0.87	31.65	1.22	22.38	36
39	1.39	20.44	0.81	35.23	1.15	24.91	39
42	1.29	22.69	0.75	39.07	1.07	27.63	42
45	1.18	25.12	0.70	43.22	0.98	30.56	45
48	1.08	27.78	0.63	47.74	0.90	33.76	48
51	0.96	30.74	0.57	52.74	0.81	37.29	51
54	0.84	34.07	0.50	58.35	0.71	41.26	54
57	0.72	37.93	0.43	64.79	0.61	45.81	57
60	0.59	42.58	0.36	72.43	0.51	51.22	60
63	0.44	48.56	0.28	82.04	0.39	58.01	63
66	0.27	57.48	0.19	95.53	0.26	67.55	66

Table A2.5 Theoretical drop data

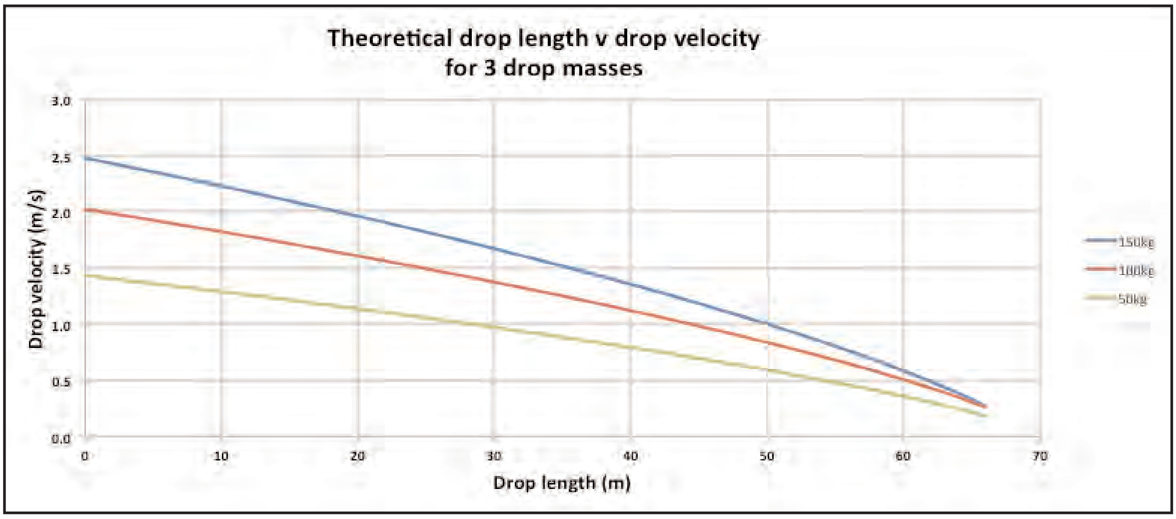


figure A2.7- Theoretical drop - 3 masses length v velocity

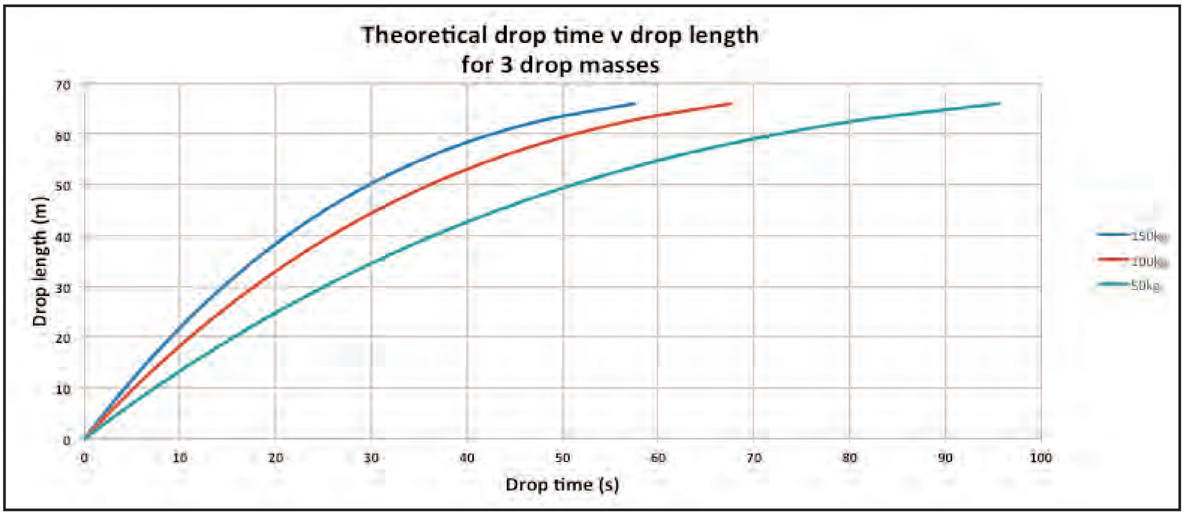


figure A2.8- Theoretical drop - 3 masses time v length

e speed (r	e torque (	e speed (r	e torque (	e speed (r	e torque (Nm)		Theo4L	Theo2+2	Theo4T	e speed (r	e torque (	Theo4S
eading shoes		2 leading 2 trailing		4 trailing shoes						4 sliding shoes		
0.1	0	0.1	0	0.1	0		0.0	0.0	0.0	0.1	0	0.0
0.1	0	0.1	0	0.1	0		0.0	0.0	0.0	0.1	0	0.0
0.1	0	0.1	0	0.1	0		0.0	0.0	0.0	0.1	0	0.0
0.1	0	0.1	0	0.1	0.3		0.0	0.0	0.0	0.1	0	0.0
0.1	0.9	0.9	0.4	0.8	0.3		0.0	0.0	0.0	0.1	0	0.0
1.6	0.5	2.6	0.2	3.9	0.4		0.0	0.1	0.1	0.1	0.2	0.0
3.5	0.8	4	0.4	5.1	0.7		0.2	0.2	0.2	1.5	0.2	0.0
6	0.8	5.2	0.6	6.7	0.6		0.6	0.3	0.3	3.9	0.3	0.2
7.3	1	7.8	0.8	8	1		0.9	0.7	0.4	5.5	0.5	0.4
9.1	1.7	9.1	1	10.5	1.2		1.4	0.9	0.7	6.9	1	0.6
10.5	1.7	10.3	1.3	11.9	1.5		1.8	1.2	0.9	8.3	1	0.8
10.5	2.3	11.7	1.7	13.2	1.9		1.8	1.6	1.1	9.6	1.2	1.1
13.3	3.1	14.2	2.1	15.6	2.1		2.9	2.3	1.5	12.4	2.1	1.9
15	3.7	15.5	2.6	16.7	2.5		3.7	2.7	1.8	13.7	1.9	2.3
17.5	4.4	16.7	3.1	18	3.1		5.0	3.2	2.0	14.8	3.4	2.7
19	5.1	19.3	3.6	20.5	3.4		5.9	4.2	2.6	16.1	3.8	3.2
20.3	6.1	20.5	4.2	21.6	3.7		6.8	4.8	2.9	18.8	4.3	4.3
21.7	6.9	21.8	5	23.1	4.4		7.7	5.4	3.4	20.1	4.8	5.0
21.7	6.9	23.1	5.6	24.1	4.9		7.7	6.0	3.6	21.4	5.9	5.6
24	8.6	25.6	6.5	25.6	5.4		9.4	7.4	4.1	23.1	7.5	6.6
26.7	9.5	27	6.8	28	6		11.7	8.3	4.9	24	8	7.1
27.5	11.8	28	7.5	29.4	6.4		12.4	8.9	5.4	26.3	9.2	8.5
28.6	12.9	29.4	9.2	30.8	7.9		13.4	9.8	6.0	27.4	10.5	9.2
29.2	14.1	30.4	9.6	31.7	8.3		14.0	10.5	6.3	28.5	12.5	10.0
29.7	15.1	32.5	10.6	33	9		14.4	12.0	6.8	29.2	12.5	10.5
30	14.9	33.3	11	35	9.4		14.7	12.6	7.7	30.3	15.2	11.3
30.3	14.5	34	11.9	36.1	9.9		15.0	13.1	8.2	31.1	15.5	11.9
31	14.5	34.8	11.5	36.9	10.3		15.7	13.7	8.6	31.3	15.7	12.1
31	14	35.3	11.8	37.8	10.1		15.7	14.1	9.0	30.8	15.9	11.7
31.2	14	35.2	11.5				15.9	14.0		30.8	16.1	11.7

Table A2.6 -Brake speed v torque comparison sliding and pivoting data

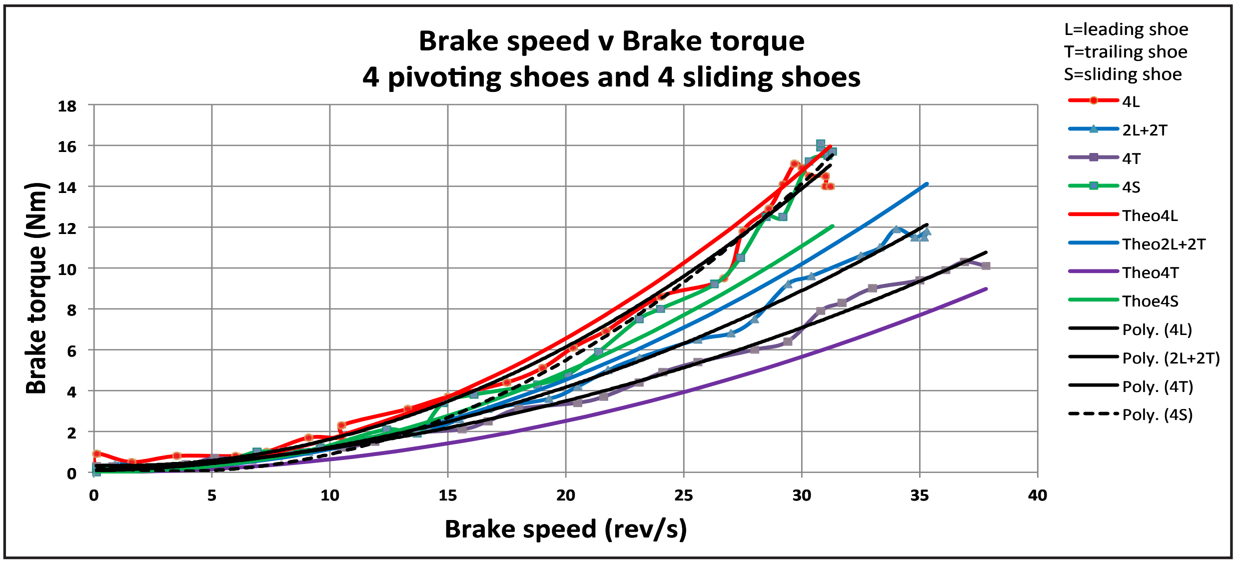


figure A2.9- Brake speed v torque comparison sliding and pivoting

Time	Brake speed (rev/s)	Brake torque (Nm)	Brake power (kw)	Theo torque (Nm)
ms				
1	0.1	0	0.0	0.0
403	0.1	0	0.0	0.0
886	0.1	0	0.0	0.0
1202	0.1	0	0.0	0.0
1601	0.1	0	0.0	0.0
2041	0.1	0.2	0.0	0.0
2400	1.5	0.2	0.0	0.0
2801	3.9	0.3	0.0	0.2
3203	5.5	0.5	0.0	0.4
3726	6.9	1	0.0	0.6
4038	8.3	1	0.1	0.8
4401	9.6	1.2	0.1	1.1
4860	12.4	2.1	0.2	1.9
5200	13.7	1.9	0.2	2.3
5629	14.8	3.4	0.3	2.7
6000	16.1	3.8	0.4	3.2
6451	18.8	4.3	0.5	4.3
6803	20.1	4.8	0.6	5.0
7376	21.4	5.9	0.8	5.6
7688	23.1	7.5	1.1	6.6
8000	24	8	1.2	7.1
8448	26.3	9.2	1.5	8.5
8800	27.4	10.5	1.8	9.2
9248	28.5	12.5	2.2	10.0
9623	29.2	12.5	2.3	10.5
10009	30.3	15.2	2.9	11.3
10402	31.1	15.5	3.0	11.9
10882	31.3	15.7	3.1	12.1
11202	30.8	15.9	3.1	11.7
11619	30.8	16.1	3.1	11.7

Table A2.7 -Brake speed v power and torque sliding shoe

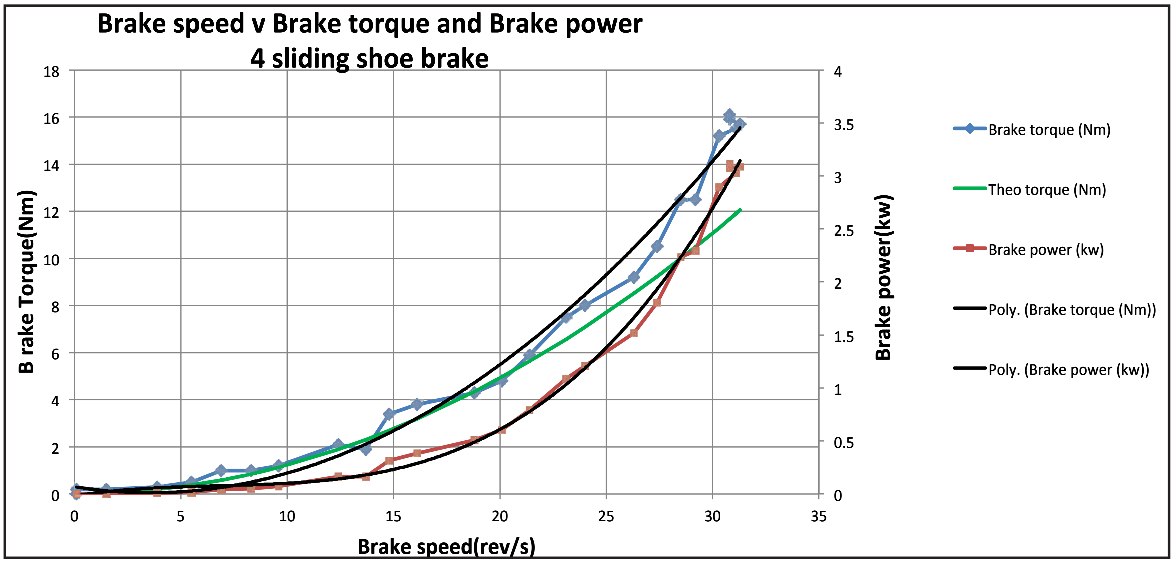


figure A2.10- Brake speed v power and torque sliding shoe

al injuries to workers by kind of accident											
	1992/93	1993/94	1994/95	1995/96	1996/97	1997/98	1998/99	1999/00	2000/01	2001/02	2002/03
Falls from height	90	81	79	64	88	92	80	68	74	69	49
slip trip fall same level	51	46	45	42	43	45	48	34	64	39	39
struck by moving object	45	33	39	32	7	41	41	35	51	46	30
Trapped by collapse	36	52	33	41	16	25	15	16	40	8	11
Falls from height	3741	3503	3552	3530	5023	5382	5454	5500	5286	4066	3880
slip trip fall same level	513	5962	5941	5800	5862	8671	9007	9087	9054	10268	10458
struck by moving object	2013	2010	2046	1978	4606	4739	4287	4370	3988	4016	3892
handling/lifting/carrying	1092	1087	1235	1134	2745	3002	2894	2862	2695	2948	3551
struck by moving vehicle	565	524	574	572	903	915	928	959	823	733	653

Table A2.8 -HSE trend data

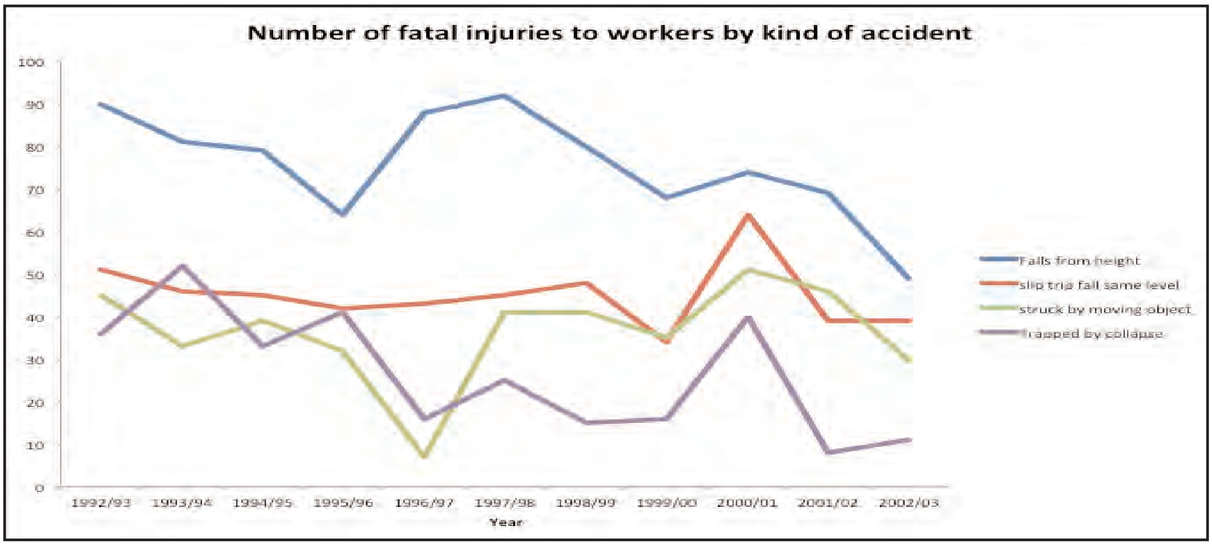


figure A2.11 - HSE - fatal injuries trend

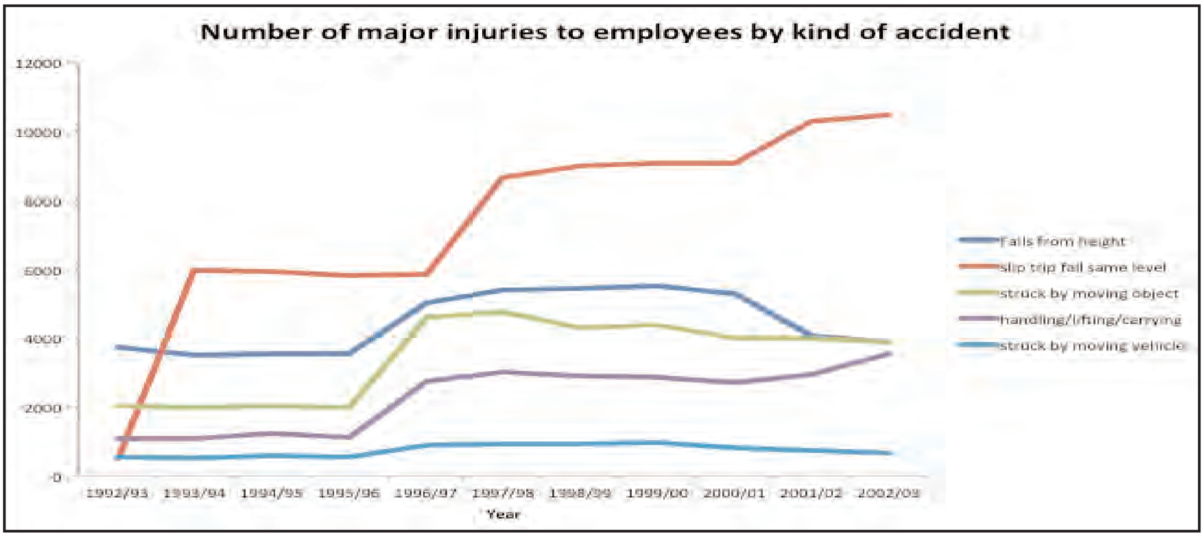


figure A2.12- HSE - major injuries trend

rain = extension over test length																	
= stress over strain in N/mm2																	
ress = force over tape width x thickess in N/mm2																	
Material	Kevlar																
Type	22 x 0.4																
Test piece	200 mm		Extension inch	0.025	0.050	0.075	0.100	0.125	0.150	0.175	0.200	0.225	0.250	0.275	0.300	0.325	0.350
Clamped	yes			0.635	1.27	1.905	2.54	3.175	3.81	4.445	5.08	5.715	6.35	6.985	7.62	8.255	8.89
Printout	1		No.1	0.150	0.175	0.300	0.450	0.620	0.825	1.000	1.120	1.400	1.625	1.850	2.100	2.250	2.500
Break start	2.2kN																
E value	7.73E3 N/mm2	A = 1kN 0.175 ext															
Stress A - B	96.6 N/mm2	B = 1.85 N 0.275 ext															
Strain A - B	1.25E-02																

Table A2.9 -Kevlar 22 x 0.4 results

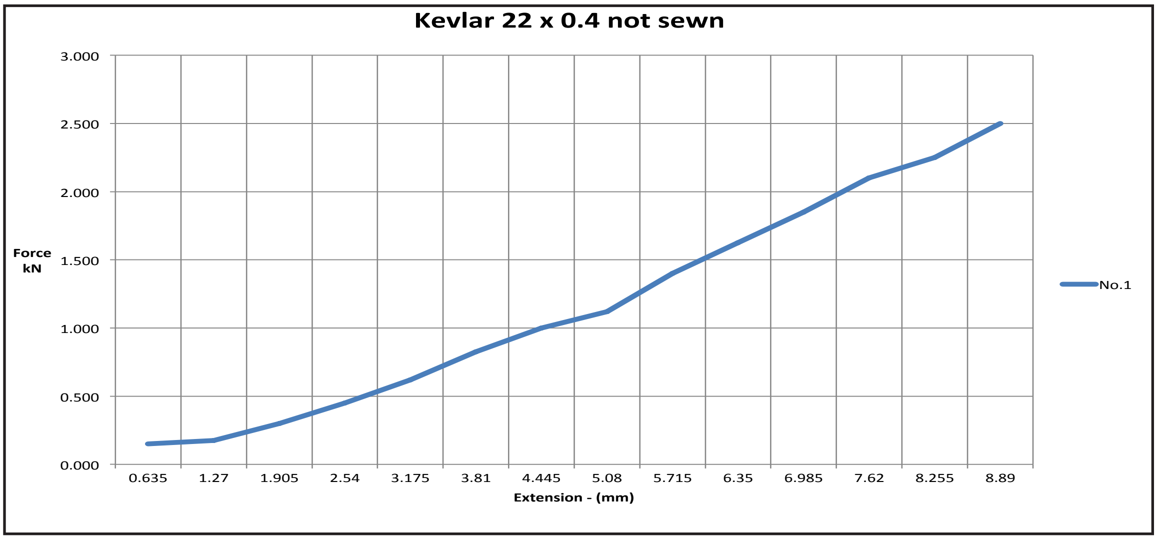


figure A2.13- kevlar 22 x 0.4

Material	RIGID Link	to compensate for Load beam and gripper deflection						
Type								
Test piece		Extension inch	0.025	0.050	0.075	0.100	0.125	
Clamped			0.635	1.27	1.905	2.54	3.175	
Printout	2	Rigid link	2.80	6.00	10.40	14.40	18.00	

Table A2.10 Rigid link

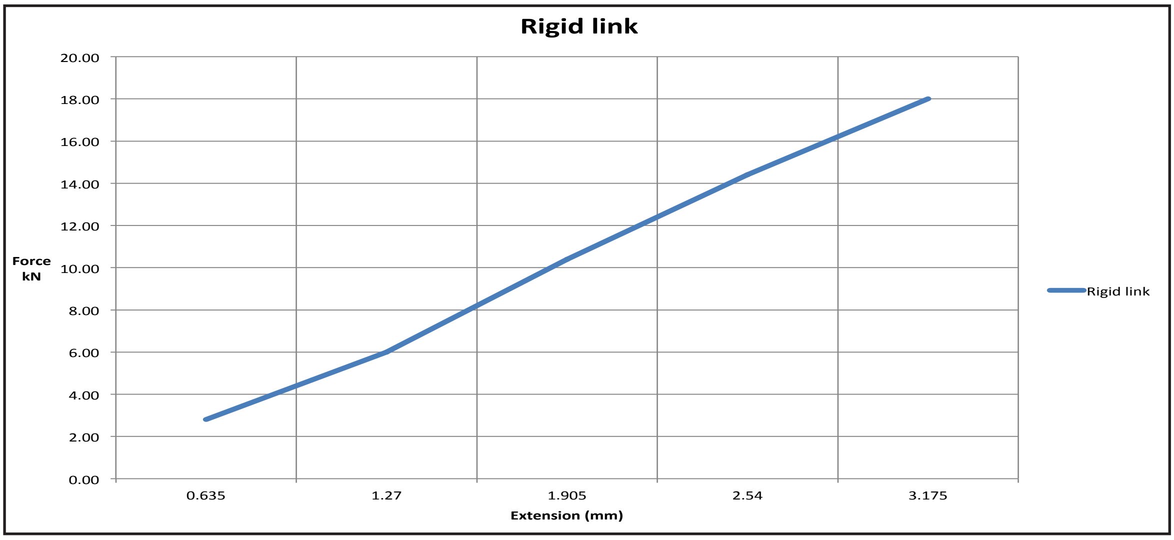


figure A2.14 - Rigid link

Material	Kevlar															
Type	25 x 0.65															
Test piece	100 mm		Extension inch	0.025	0.050	0.075	0.100	0.125	0.150	0.175	0.200	0.225	0.250	0.275	0.298	
Clamped	yes		Extension mm	0.635	1.27	1.905	2.54	3.175	3.81	4.445	5.08	5.715	6.35	6.985	7.5565	
Printout	240 - 1		240-1	0.9	2.4	4	5.5	7.2	8.8	10.5	12.3	14	15.7	17.4	19	
Break start	ak suddenly at 19 kN		240-2	0.7	2	3.4	4.8	6.6	8.3	10	11.7	13.6	15.4	17.1	18.6	
E value	28.49 E3 N/mm2		264-1	0.6	1.8	3	4.5	6	7.8	9.6	11.3	13.1	15	16.7	18.4	19
Stress A - B	615.4 N/mm2		264-2	0.8	1.8	3.2	4.5	6	7.6	9.4	11.2	13	14.8	16.5	18.2	18.2
Strain A - B	21.6 E-3															
5kN 0.1075 ext																
16 N 0.245 ext																

Table A2.11 - Kevlar 25 not sewn

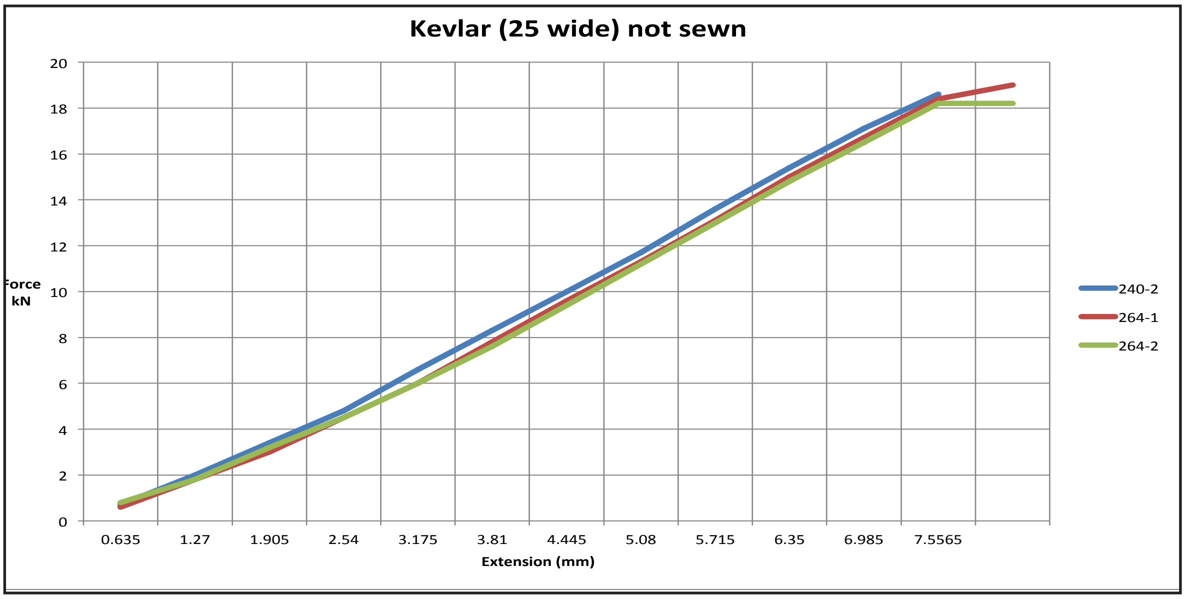


figure A2.15 - kevlar 25 not sewn

Material	Kevlar																					
Type	22 x 0.45																					
Test piece	200 mm		Extension inch	0.025	0.050	0.075	0.100	0.125	0.150	0.175	0.200	0.225	0.250	0.275	0.300	0.325	0.350	0.375	0.400	0.425	0.450	0.455
Clamped	yes		Extension mm	0.635	1.27	1.905	2.54	3.175	3.81	4.445	5.08	5.715	6.35	6.985	7.62	8.255	8.89	9.525	10.16	10.795	11.43	11.557
Printout	No.6		No.6	0.15	0.25	0.35	0.55	0.75	1	1.2	1.5	1.8	2.1	2.4	2.7	3.05	3.38	3.7	4	4.35	4.6	4.7
Break start	2.2kN		No.5	0.1	0.2	0.35	0.55	0.8	1	1.3	1.55	1.9	2.2	2.5	2.8	3.15	3.45	3.8	4.15	4.5	4.6	
E value	9.7 E3 N/mm2		No.4	0.8	2.2	3.8	5.6	7.6	9.4	10.3												
Stress A - B	121 N/mm2		No.3	1	2.6	4.4	6.6	8.9	10.6													
Strain A - B	12.5 E3																					

Table A2.12 -Kevlar 20 and 22 wide not sewn

figure A2.16 - kevlar 20 and 22 wide not sewn

Material	264 kevlar												
Type	vide 16 mm nylon sq												
Test piece	35 mm		Extension inch	0.050	0.100	0.150	0.200	0.250	0.300	0.350	0.400	0.450	0.460
Clamped	yes			1.27	2.54	3.81	5.08	6.35	7.62	8.89	10.16	11.43	11.684
Printout	12		12 S3	0.2	0.6	1.2	2.1	3.1	4.3	5.6	7	8.4	8.6
Break start			12 S2	0.3	0.8	1.4	2.3	3.2	4.5	5.6	7.1	8.6	
E value			12 S1	0.3	0.7	1.2	2.2	3.2	4.4	5.9	7.4	8.9	9.6
Stress A - B			11 - S1	0.3	1	1.4	2.5	3.6	4.9	6.4	7.7	7.9	
Strain A - B			11 - S2	0.3	0.8	1.5	2.5	3.6	4.9	6.3	7.7	7.8	
			11 - S3	0.3	0.8	1.6	2.4	3.4	4.7	6	7.4	7.6	

Table A2.13 kevlar with nylon results

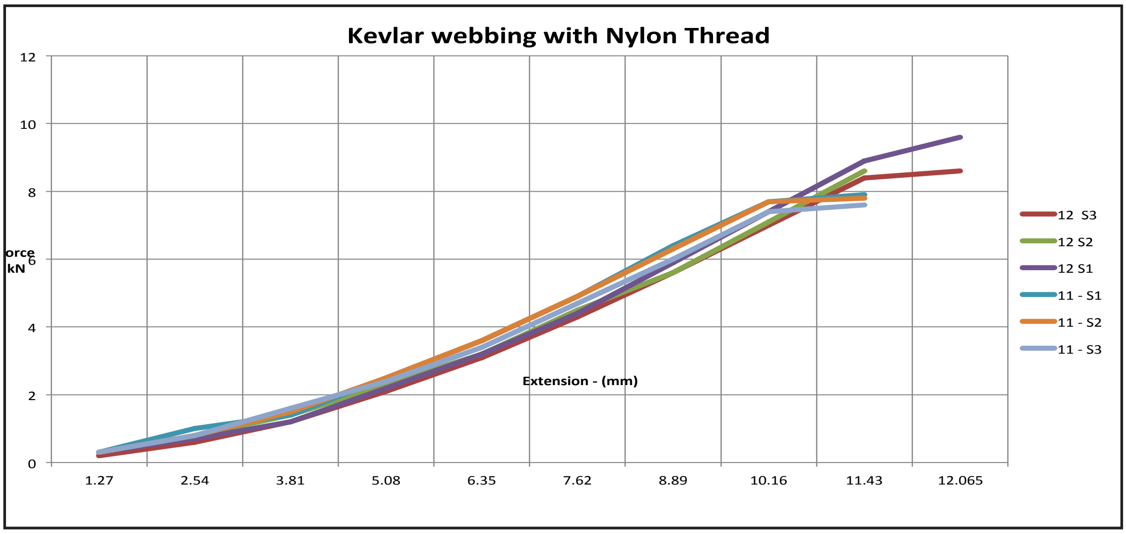


figure A2.17- kevlar with nylon

Material	Kevlar																												
Type																													
Test piece			Extension inch																										
Clamped	yes			1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23			
Printout	A10.1		A10.1	0.6	0.8	0.9	1	1.2	1.4	1.5	1.5	1.7	1.8	2	2.2	2.4	2.5	2.9	3.1	3.3	3.5	3.7	3.9	4	4.2	4.4			
Break start			A10.2	0.5	0.6	0.7	0.8	0.9	1	1.1	1.2	1.3	1.4	1.5	1.7	1.8	2	2.3	2.5	2.7	2.8	3	3.2	3.2	3.5	3.7	4	4.2	4.5
E value			A15.1	0.6	0.7	0.8	0.9	1	1.1	1.2	1.2	1.4	1.5	1.6	1.8	1.9	2.1	2.4	2.6	2.8	3	3.2							
Stress A - B			A15.2	0.5	0.6	0.7	0.8	0.9	1	1.1	1.1	1.3	1.4	1.5	1.7	1.8	2	2.3	2.5	2.7	2.8	3	3.2	3.2	3.3				
Strain A - B			B10.1	0.6	0.7	1	1.2	1.4	1.7	1.9	1.9	2.3	2.6	3	3.4	3.7	4.1	4.8	5.3	5.7	6.2	6.6	7.1	7.2	7.7	8.2	8.7	9	
			B15.1	0.5	0.7	0.9	1	1.3	1.5	1.7	1.7	2.1	2.4	2.7	3.2	3.5	3.7	4.6	5	5.5	5.9	6.3	6.7	6.7	7.3	7.8	8.3	8.4	
			B15.2	0.7	0.8	1	1.2	1.3	1.5	1.8	1.8	2.2	2.5	2.8	3.2	3.5	4	4.7	5.1	5.5	6	6.5	7	7	7.5				

Table A2.14 -kevlar with kevlar results

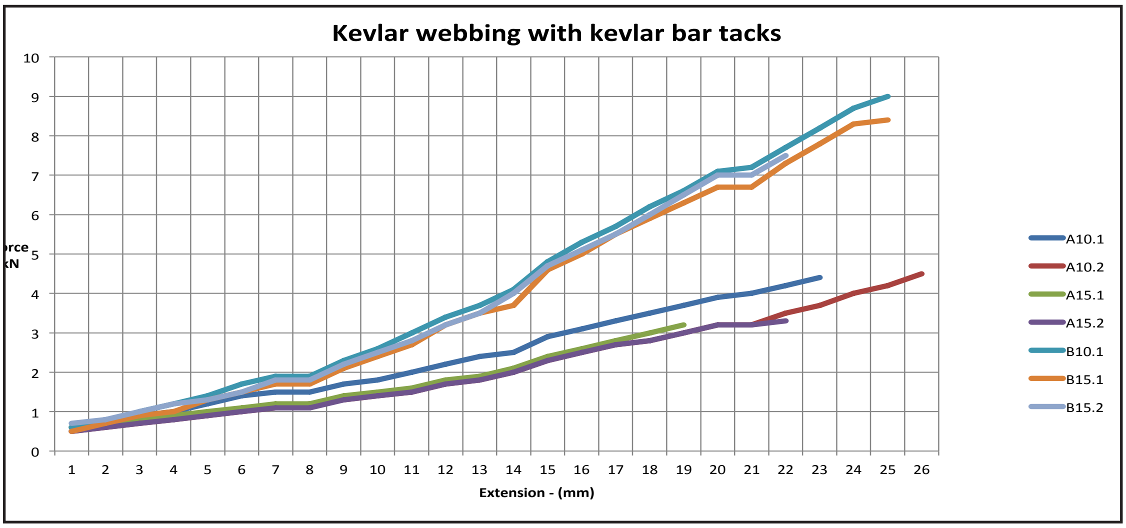


figure A2.18 -kevlar with kevlar

Material	Kevlar												
Type													
Test piece		Extension inch											
Clamped	yes	Extension mm	1	2	3	4	5	6	7	8	9	9.7	
Printout	K22B	K22B	0.9	1.6	2.6	4	5.7	7.7	9.7	11.8	14.3	15.8	
Sheet 21													
Material	Kevlar												
Type													
Test piece		Extension inch											
Clamped	yes	Extension mm	1	2	3	4	5	6	7	8	9	9.8	
Printout	K22A	K22A	0.9	1.6	2.8	4.2	6.1	8.1	10.2	12.5	15	16.5	
Material	Kevlar												
Type													
Test piece		Extension inch											
Clamped	yes	Extension mm	1	2	3	4	5	6	7	8	8.8		
Printout	K21B	K21B	0.9	1.8	3.3	5.2	7.5	10	12.8	15.9	18.6		
Break start	No										off the meter at 18.6		
Material	Kevlar												
Type													
Test piece		Extension inch											
Clamped	yes	Extension mm	1	2	3	4	5	6	7	8	9		
Printout	K21A	K21A	1.2	2.1	3.6	5.3	7.5	9.8	12.4	15.2	18.2		
Break start	No										off the meter		
Material	Kevlar												
Type													
Test piece		Extension inch											
Clamped	yes	Extension mm	1	2	3	4	5	6	7	8	9	10	11
Printout	K23A	K23A	0.7	1.1	1.5	2.1	2.4	3.6	4.6	5.6	6.8	8	
Break start													

Table A2.15 -Kevlar with zylon results

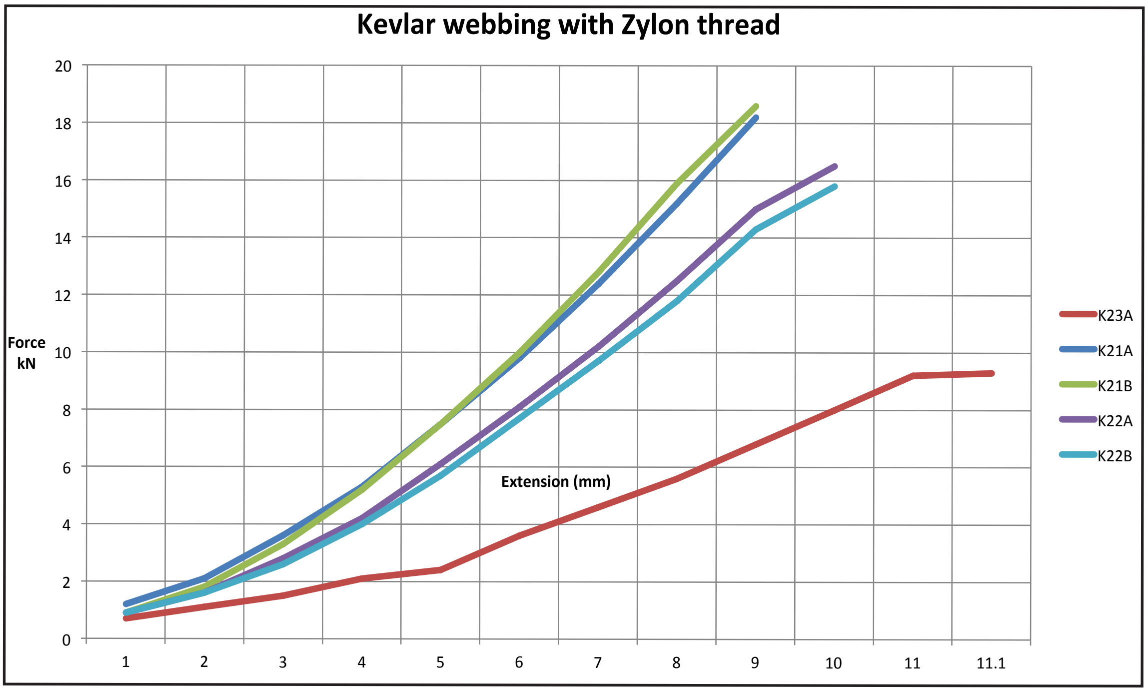


figure A2.19 - Kevlar with zylon

Table A2.16 -Tower - 40 kg drop

Drop length(m)	Drop velocity (m/s)	Theoretical velocity e=.95	Theoritical velocity e=.65
12.2	1.16	1.42	1.12
12.7	1.09	1.41	1.11
13	1.06	1.40	1.11
13.7	1.06	1.39	1.10
14.1	1.06	1.38	1.10
14.9	1.06	1.37	1.08
15.5	0.99	1.36	1.08
15.8	0.96	1.35	1.07
16.5	0.94	1.34	1.06
16.8	0.99	1.33	1.06
17.4	1	1.32	1.05
17.8	1.01	1.31	1.04
18.4	0.97	1.30	1.04
18.7	0.96	1.30	1.03
19.4	0.94	1.28	1.02
19.7	0.93	1.28	1.02
20.1	0.92	1.27	1.01
20.9	0.92	1.25	1.00
21.3	0.93	1.25	0.99
21.8	0.93	1.24	0.99
22.2	0.91	1.23	0.98
22.9	0.9	1.22	0.97
23.3	0.89	1.21	0.96
23.5	0.88	1.20	0.96
24.1	0.88	1.19	0.95
24.7	0.88	1.18	0.94
25	0.88	1.17	0.94
25.5	0.88	1.16	0.93
25.8	0.88	1.16	0.93
26.4	0.85	1.15	0.92
26.9	0.85	1.14	0.91
27.2	0.85	1.13	0.91
27.7	0.84	1.12	0.90
28	0.85	1.11	0.89
28.5	0.82	1.10	0.89
29	0.84	1.09	0.88
29.4	0.83	1.09	0.87
29.9	0.82	1.08	0.86
30	0.82	1.07	0.86
30.6	0.79	1.06	0.85
30.8	0.8	1.06	0.85
31.3	0.77	1.05	0.84
31.6	0.78	1.04	0.84
32.3	0.77	1.03	0.83
32.6	0.74	1.02	0.82
32.9	0.75	1.01	0.82
33.1	0.74	1.01	0.81
33.6	0.76	1.00	0.81
33.9	0.73	0.99	0.80
34.7	0.72	0.98	0.79
34.8	0.7	0.97	0.79
35.4	0.71	0.96	0.78
35.7	0.71	0.95	0.77
36	0.71	0.95	0.77
36.2	0.72	0.94	0.76
36.8	0.7	0.93	0.75
37.2	0.69	0.92	0.75
37.2	0.69	0.92	0.75
37.8	0.69	0.91	0.74
38.1	0.69	0.90	0.73
38.7	0.66	0.89	0.72
38.9	0.67	0.89	0.72
39.1	0.66	0.88	0.72
39.2	0.66	0.88	0.71
40	0.67	0.86	0.70
40	0.67	0.86	0.70
40.7	0.67	0.85	0.69
40.5	0.67	0.85	0.69
41.4	0.64	0.83	0.68
41.5	0.62	0.83	0.67
41.7	0.63	0.82	0.67
41.9	0.64	0.82	0.67
42.7	0.61	0.80	0.65
42.5	0.61	0.81	0.66
43.3	0.59	0.79	0.64
43.7	0.6	0.78	0.63
43.6	0.59	0.78	0.64
43.9	0.61	0.77	0.63
44.4	0.6	0.76	0.62
45	0.57	0.75	0.61
45	0.56	0.75	0.61
45	0.58	0.75	0.61
45.4	0.58	0.74	0.60
46.1	0.57	0.72	0.59
45.6	0.55	0.73	0.60
46.3	0.55	0.72	0.59
46.3	0.53	0.72	0.59
47.2	0.55	0.70	0.57
46.8	0.53	0.71	0.58
47.2	0.51	0.70	0.57
48	0.52	0.68	0.56
48.3	0.52	0.67	0.55
48.6	0.52	0.66	0.54
48.2	0.51	0.67	0.55
49.2	0.53	0.65	0.53
48.8	0.51	0.66	0.54
49.7	0.48	0.64	0.52
49.7	0.5	0.64	0.52
50.3	0.48	0.62	0.51
50.1	0.5	0.63	0.51
50.8	0.48	0.61	0.50
50.6	0.47	0.62	0.51
50.8	0.48	0.61	0.50
50.6	0.47	0.62	0.51
50.9	0.48	0.61	0.50
51.4	0.46	0.60	0.49
51.4	0.44	0.60	0.49
52.1	0.43	0.58	0.48
51.9	0.43	0.58	0.48
52.7	0.45	0.56	0.46
52.6	0.42	0.57	0.46
53.5	0.43	0.54	0.45
53.6	0.44	0.54	0.44
53	0.43	0.56	0.46
53.4	0.41	0.55	0.45
53.6	0.42	0.54	0.44
53.7	0.41	0.54	0.44
53.6	0.39	0.54	0.44
54.4	0.41	0.52	0.43
54.1	0.38	0.53	0.43
55	0.4	0.50	0.41
55.3	0.38	0.50	0.41
55	0.37	0.50	0.41
55.4	0.41	0.49	0.41
55.2	0.36	0.50	0.41
55.9	0.35	0.48	0.40
56.4	0.36	0.47	0.38
56.8	0.38	0.46	0.38
56.9	0.36	0.45	0.37
56	0.34	0.48	0.39
56.8	0.35	0.46	0.38
56.8	0.36	0.46	0.38
57.7	0.34	0.43	0.36
56.8	0.34	0.46	0.38
57.9	0.31	0.42	0.35
57	0.35	0.45	0.37
57.2	0.34	0.44	0.37
58.3	0.36	0.41	0.34
58.6	0.35	0.41	0.34
57.7	0.31	0.43	0.36
59	0.35	0.39	0.33
58	0.31	0.42	0.35
58.2	0.32	0.42	0.34
58.5	0.31	0.41	0.34
59.8	0.31	0.37	0.31
59.1	0.29	0.39	0.32
58.8	0.29	0.40	0.33
60.3	0.3	0.36	0.30
60.4	0.29	0.35	0.29
60.2	0.33	0.36	0.30
59.6	0.26	0.38	0.31
59.8	0.27	0.37	0.31
60.9	0.26	0.34	0.28
59.9	0.28	0.37	0.30
61.4	0.26	0.32	0.27
60.2	0.23	0.36	0.30
61	0.28	0.34	0.28
61.7	0.28	0.31	0.26
61.9	0.25	0.31	0.26
62	0.28	0.30	0.25
61.6	0.22	0.32	0.26
61	0.25	0.34	0.28
61.8	0.23	0.31	0.26
62.3	0.24	0.30	0.25
61.3	0.21	0.33	0.27
62.9	0.21	0.28	0.23
63	0.26	0.27	0.23
63.1	0.27	0.27	0.22
61.8	0.2	0.31	0.26
62.2	0.23	0.30	0.25
62.1	0.2	0.30	0.25
63.3	0.24	0.26	0.22
63.8	0.26	0.25	0.21
63.5	0.24	0.26	0.21
63	0.2	0.27	0.23
64	0.21	0.24	0.20
63.3	0.18	0.26	0.22
62.9	0.19	0.28	0.23
63.6	0.2	0.25	0.21
64.1	0.21	0.24	0.20
63.4	0.19	0.26	0.22
63.3	0.18	0.26	0.22
64.7	0.24	0.22	0.18
63.8	0.19	0.25	0.21
64.8	0.18	0.21	0.18
65.2	0.22	0.20	0.17
64.4	0.19	0.23	0.19
63.9	0.17	0.24	0.20
65.5	0.22	0.19	0.16

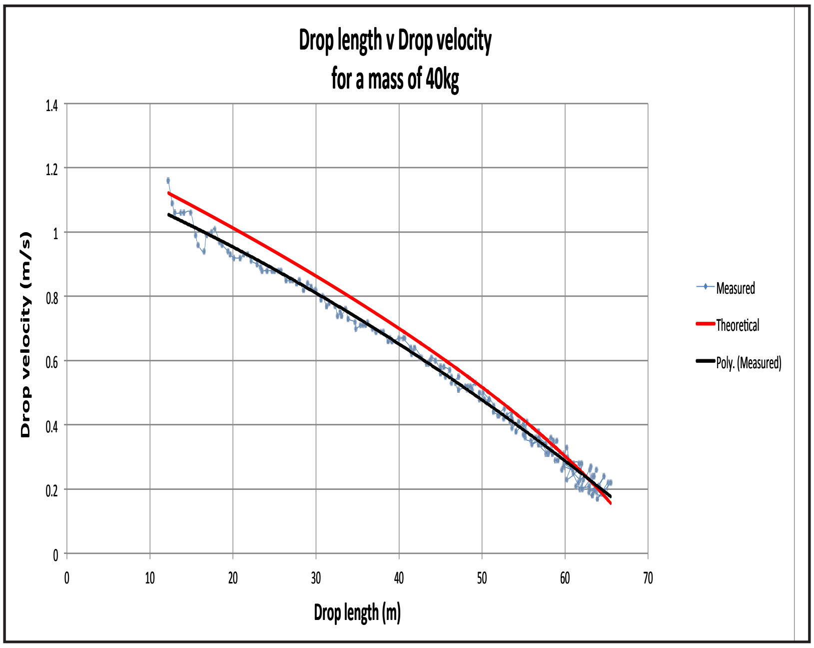


figure A2.20- Tower - 40 kg drop

Appendix 3 - Materials

Information contained in this section

Thsee have been removed

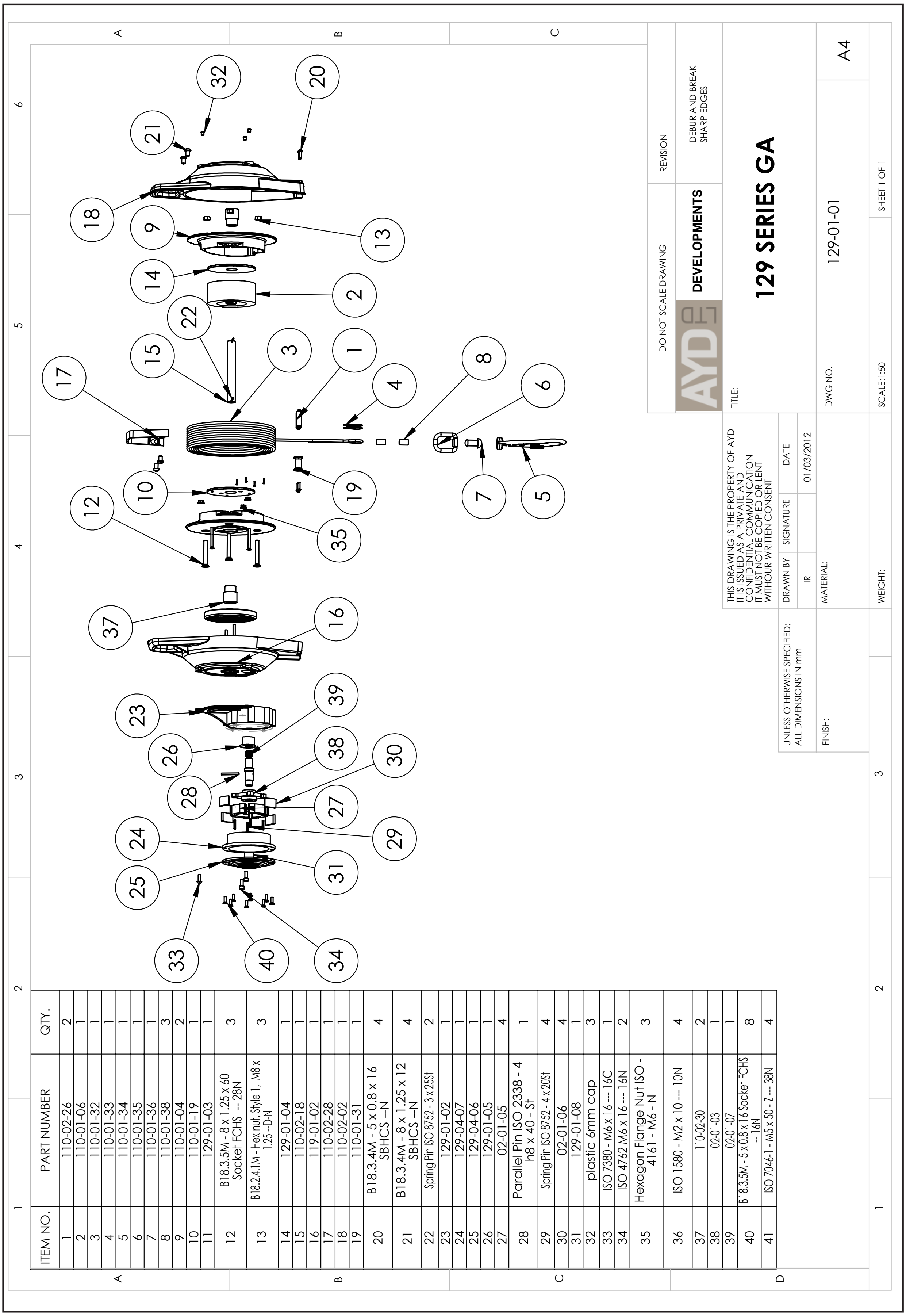
Appendix 4 - Published work

1210	Manual descender - Descent control limited
1270	Automatic reciprocating descender - Descent control Limited
1280	Automatic rewinding descender - Descent control Limited
1360	Pulley descent controller - Descent control Limited

These have been removed

Appendix 5 - Design drawings- brake example 129

129 - 01 - 01	Exploded general arrangement
129 - 01 - 02	Module housing
129 - 01 - 03	Gear ring
129 - 01 - 05	D hole bush
129 - 01 - 08	End cap bearing
129 - 01 - 09	Exploded brake general arrangement
129 - 04 - 06	End cap
129 - 04 - 07	Descent ring (lining)
02 - 01 - 01	Brake sub assembly
02 - 01 - 03	Brake hub 4 leading 4 trailing
02 - 01 - 04	Brake hub 2 leading 2 trailing
02 - 01 - 05	Brake shoe
02 - 01 - 06	Brake shoe friction pad
02 - 01 - 07	Long pinion



ITEM NO.	PART NUMBER	QTY.
1	110-02-26	2
2	110-01-06	1
3	110-01-32	1
4	110-01-33	1
5	110-01-34	1
6	110-01-35	1
7	110-01-36	1
8	110-01-38	3
9	110-01-04	2
10	110-01-19	1
11	129-01-03	1
12	B18.3.5M - 8 x 1.25 x 60 Socket FCHS -- 28N	3
13	B18.2.4.1M - Hex nut, Style 1, M8 x 1.25 --D-N	3
14	129-01-04	1
15	110-02-18	1
16	119-01-02	1
17	110-02-28	1
18	110-02-02	1
19	110-01-31	1
20	B18.3.4M - 5 x 0.8 x 16 SBHCS --N	4
21	B18.3.4M - 8 x 1.25 x 12 SBHCS --N	4
22	Spring Pin ISO 8752 - 3 x 25St	2
23	129-01-02	1
24	129-04-07	1
25	129-04-06	1
26	129-01-05	1
27	02-01-05	4
28	Parallel Pin ISO 2338 - 4 h8 x 40 - St	1
29	Spring Pin ISO 8752 - 4 x 20St	4
30	02-01-06	4
31	129-01-08	1
32	plastic 6mm cap	3
33	ISO 7380 - M6 x 16 --- 16C	1
34	ISO 4762 M6 x 16 --- 16N	2
35	Hexagon Flange Nut ISO - 4161 - M6 - N	3
36	ISO 1580 - M2 x 10 --- 10N	4
37	110-02-30	2
38	02-01-03	1
39	02-01-07	1
40	B18.3.5M - 5 x 0.8 x 16 Socket FCHS -- 16N	8
41	ISO 7046-1 - M5 x 50 - Z --- 38N	4

Figure A5.1 : Exploded general arrangement

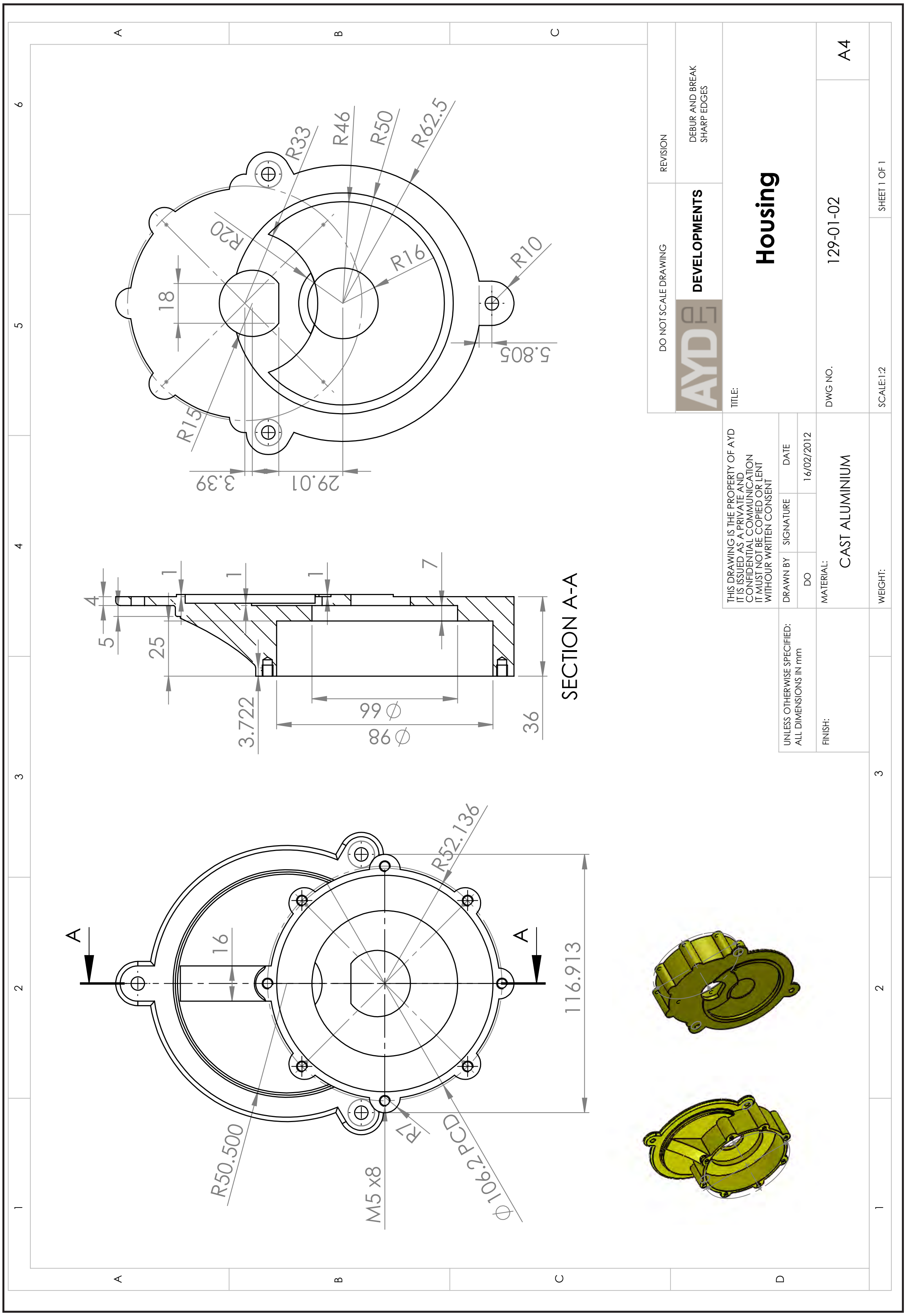


Figure A5.2 : Module housing

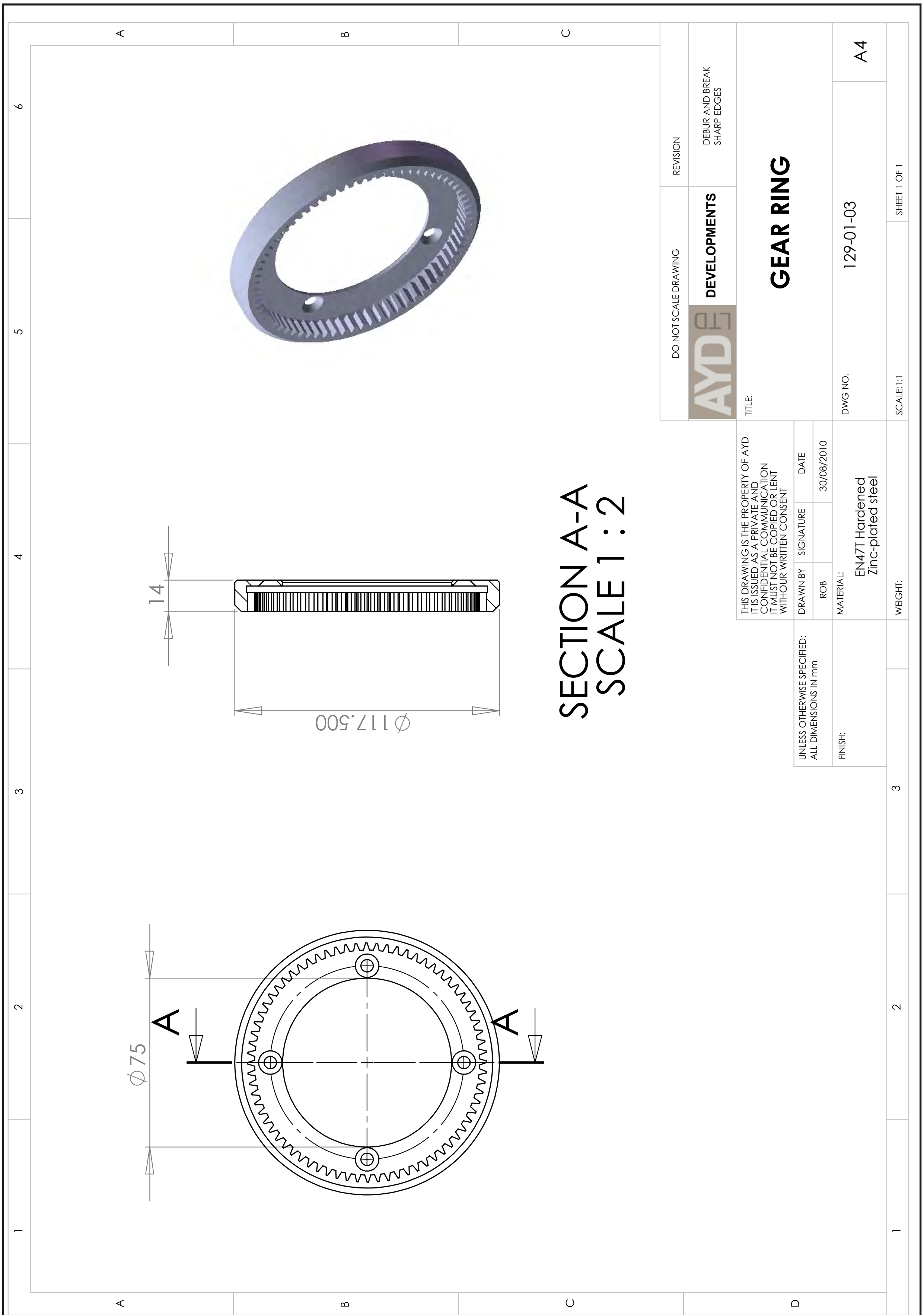


Figure A5.3 : Gear ring

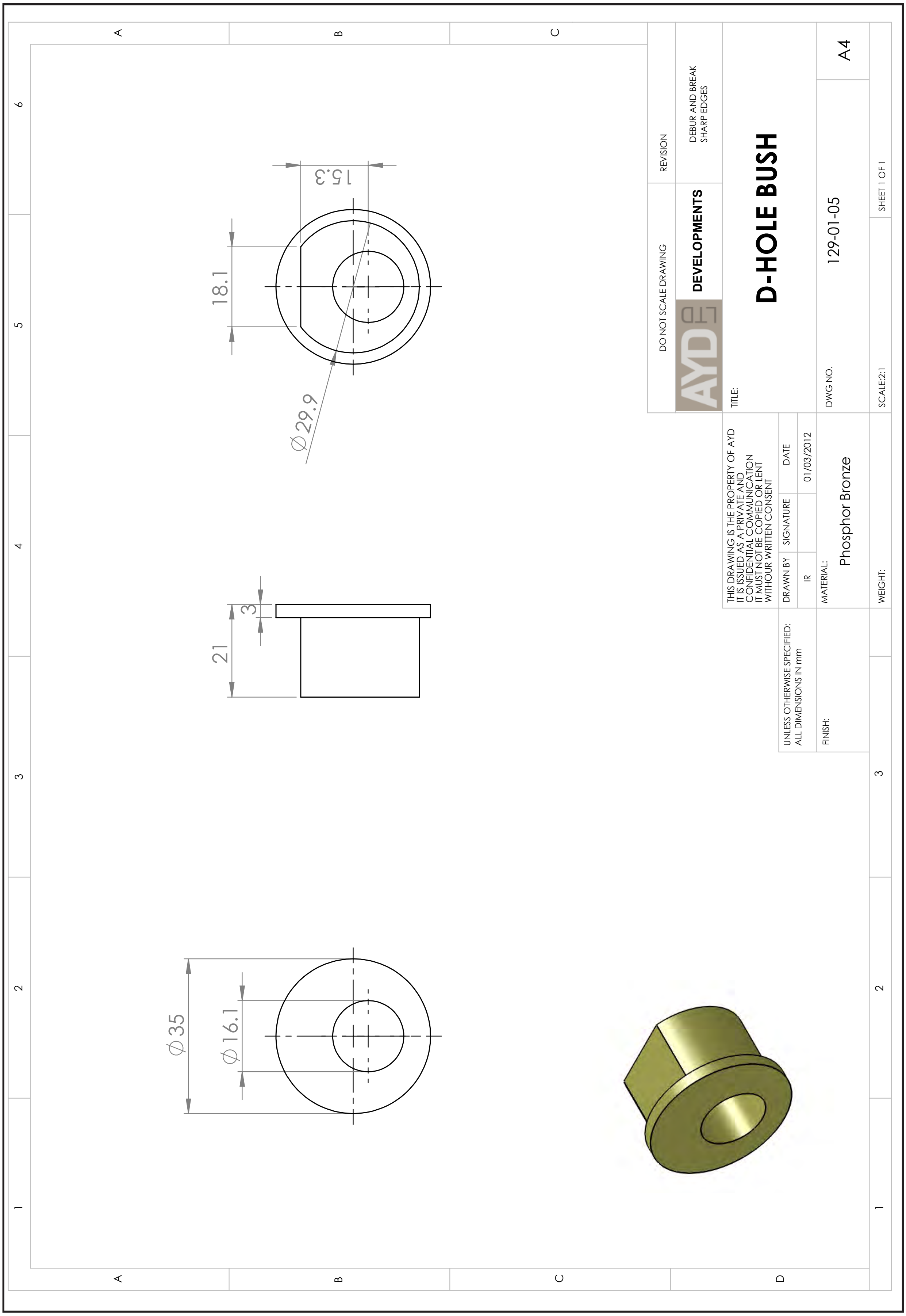


Figure A5.4 :D hole bush

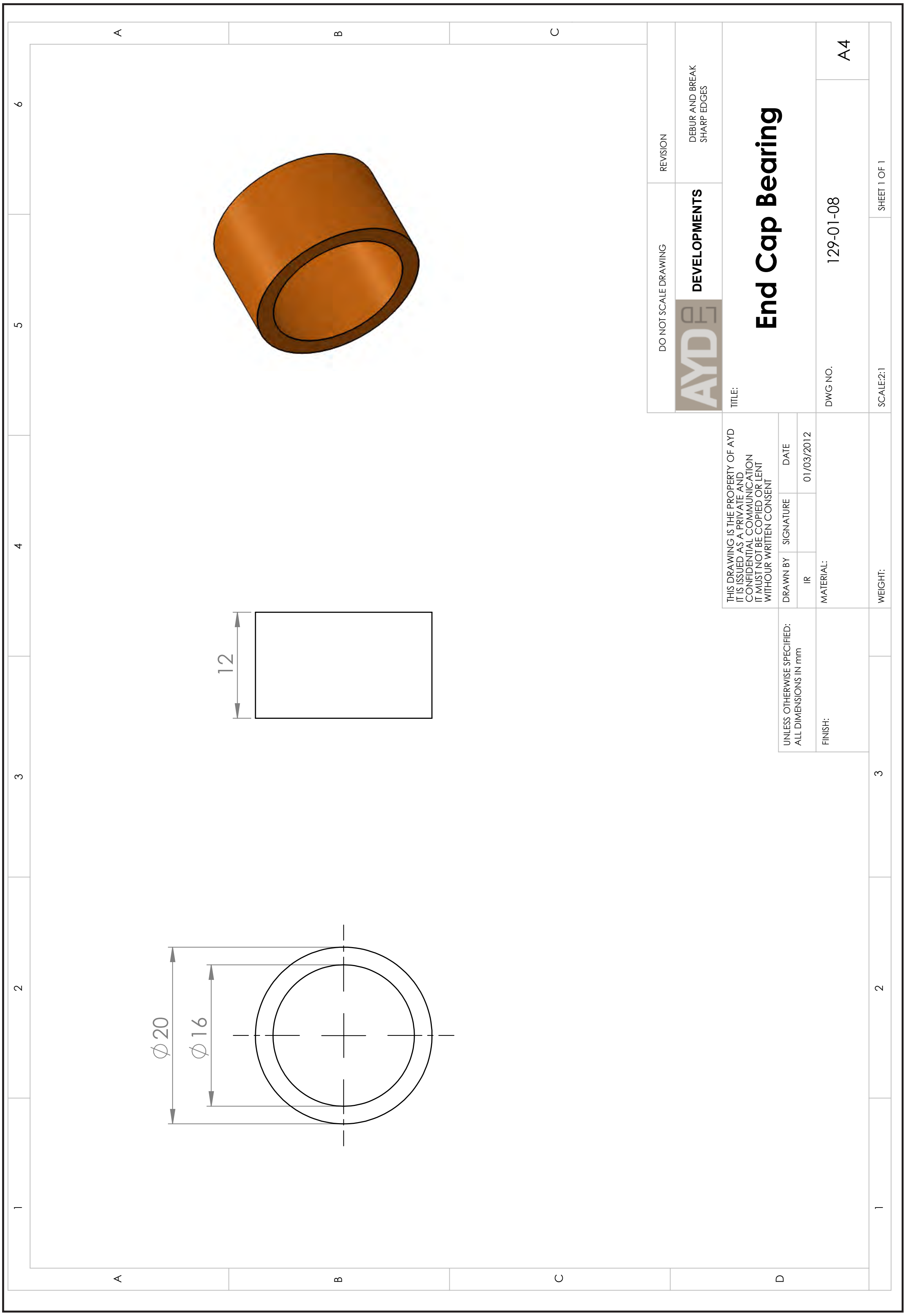


Figure A5.5 : End cap bearing

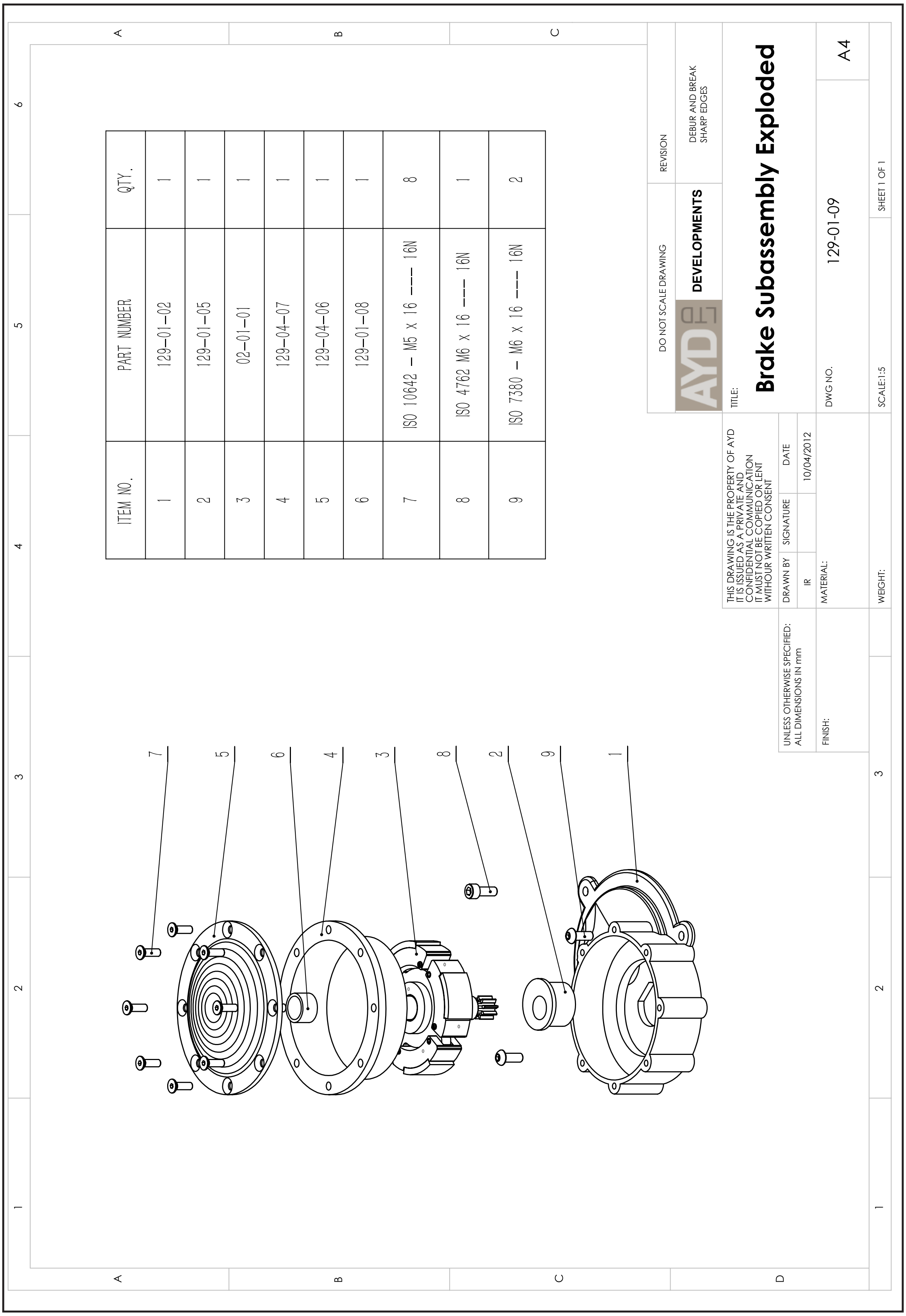


Figure A5.6 : Expolded brake general arrangement

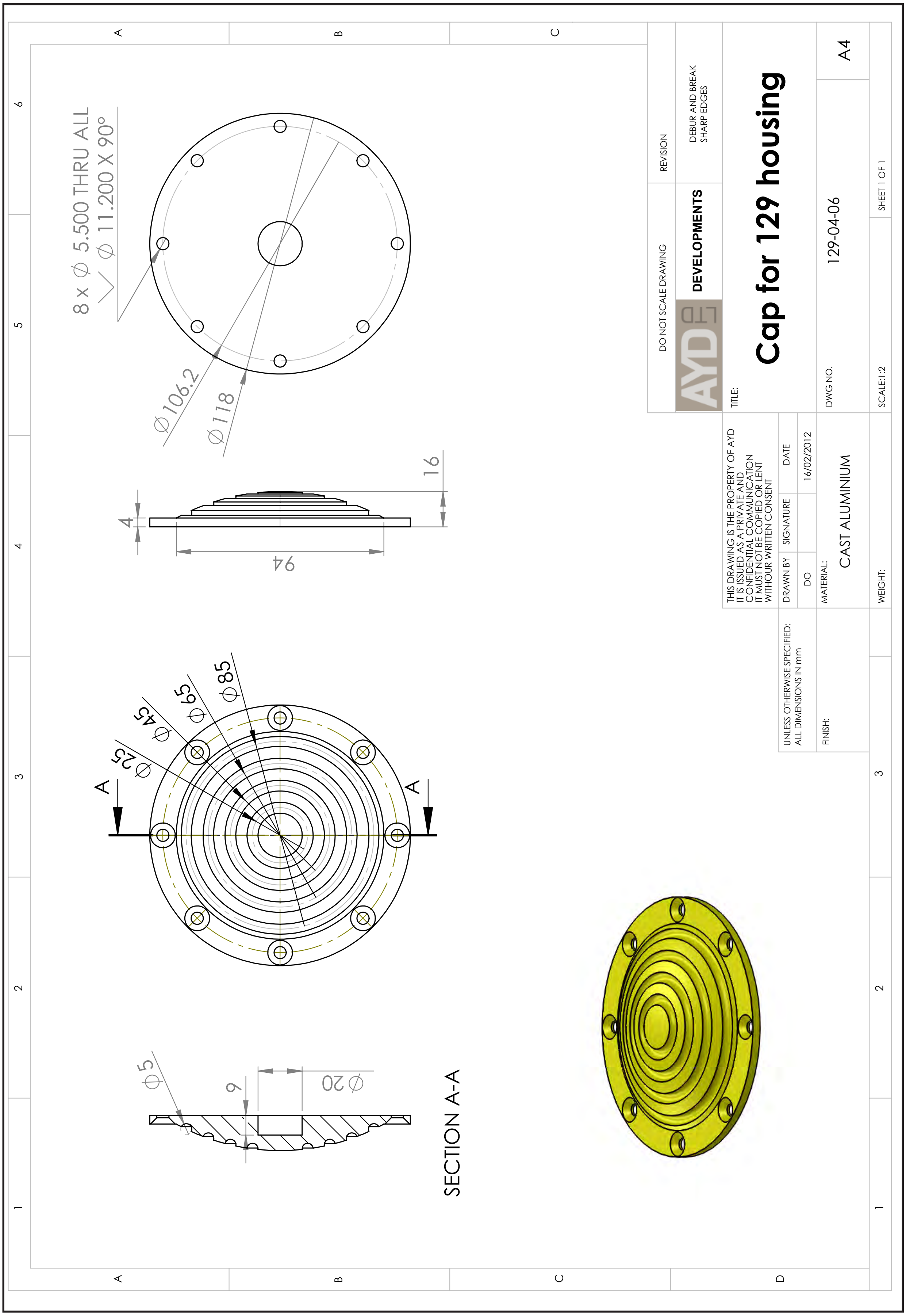


Figure A5.7 : End cap

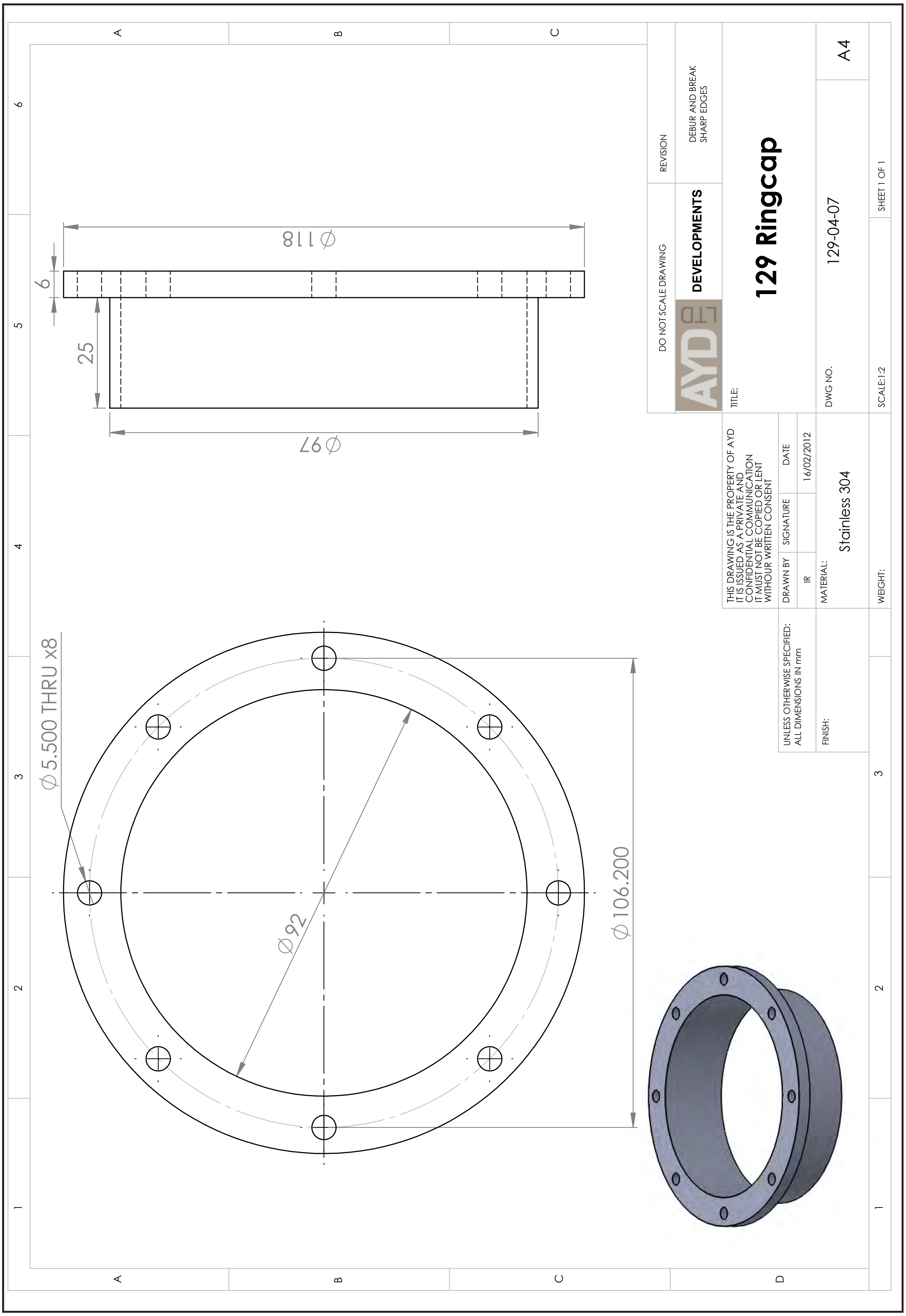
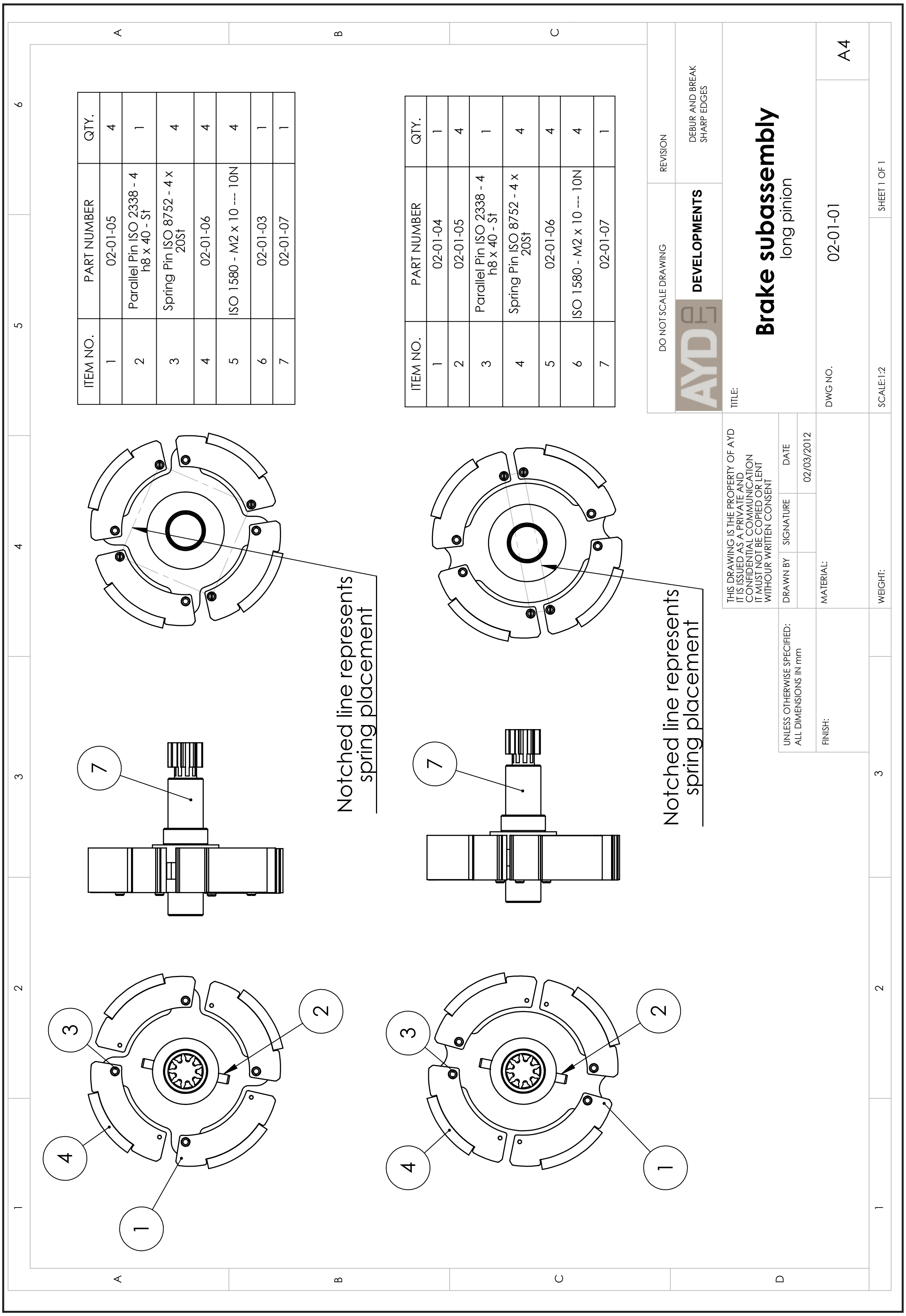


Figure A5.8: Descent ring (lining)



ITEM NO.	PART NUMBER	QTY.
1	02-01-05	4
2	Parallel Pin ISO 2338 - 4 h8 x 40 - St	1
3	Spring Pin ISO 8752 - 4 x 20St	4
4	02-01-06	4
5	ISO 1580 - M2 x 10 --- 10N	4
6	02-01-03	1
7	02-01-07	1

ITEM NO.	PART NUMBER	QTY.
1	02-01-04	1
2	02-01-05	4
3	Parallel Pin ISO 2338 - 4 h8 x 40 - St	1
4	Spring Pin ISO 8752 - 4 x 20St	4
5	02-01-06	4
6	ISO 1580 - M2 x 10 --- 10N	4
7	02-01-07	1

Figure A5.9 :Brake sub assembly

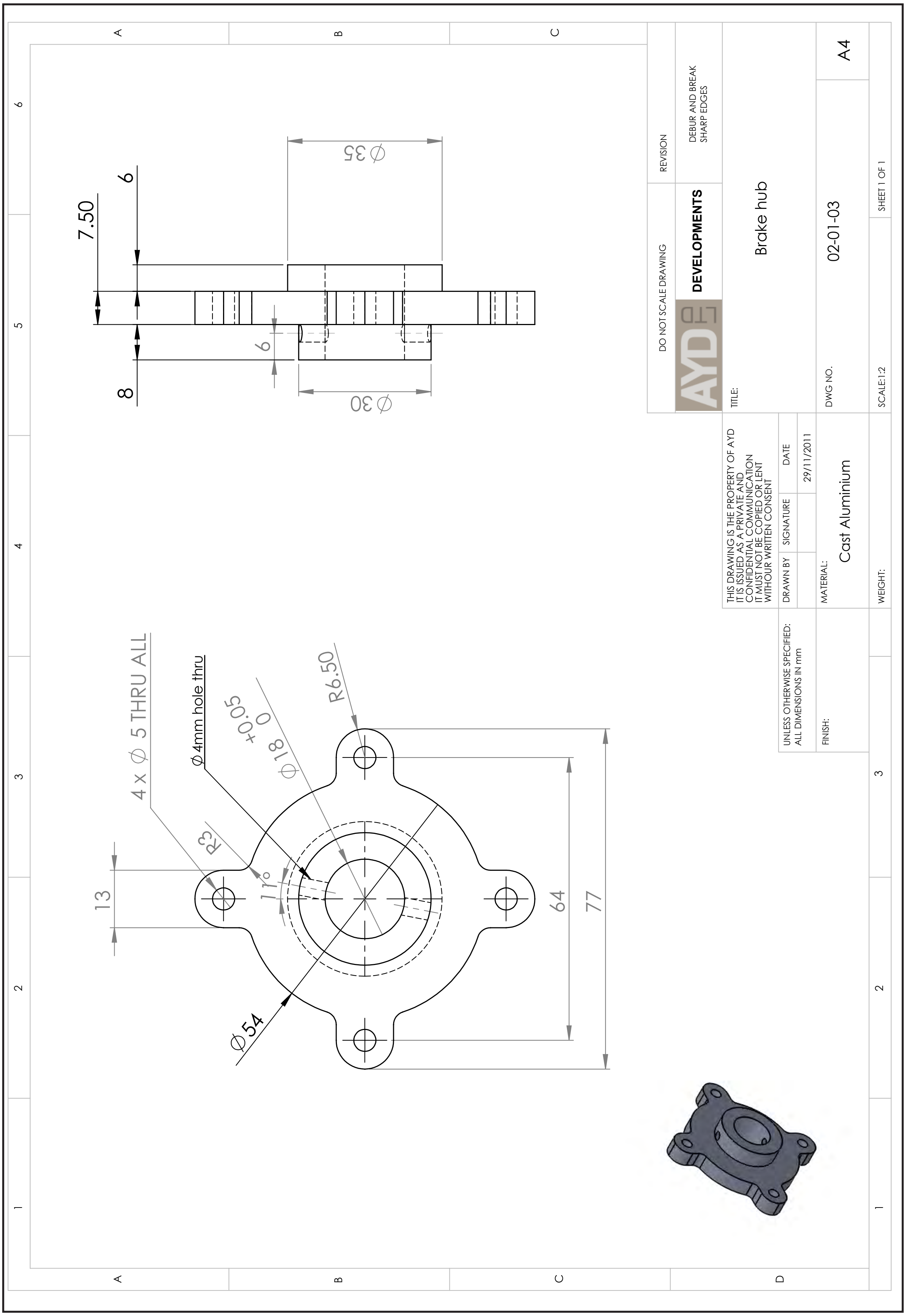


Figure A5.10 :Brake hub 4 leading 4 trailing

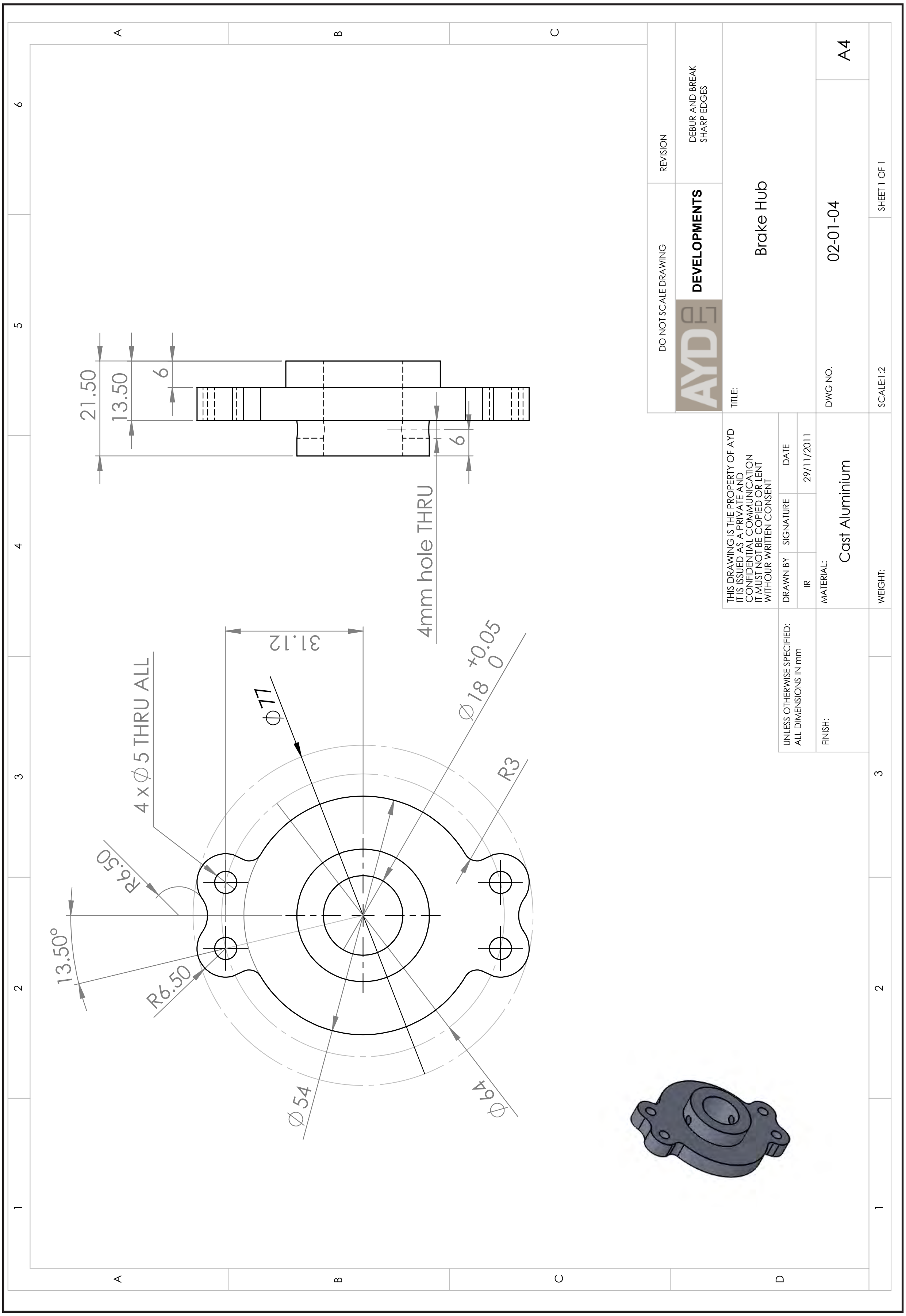


Figure A5.11 : Brake hub 2 leading 2 trailing

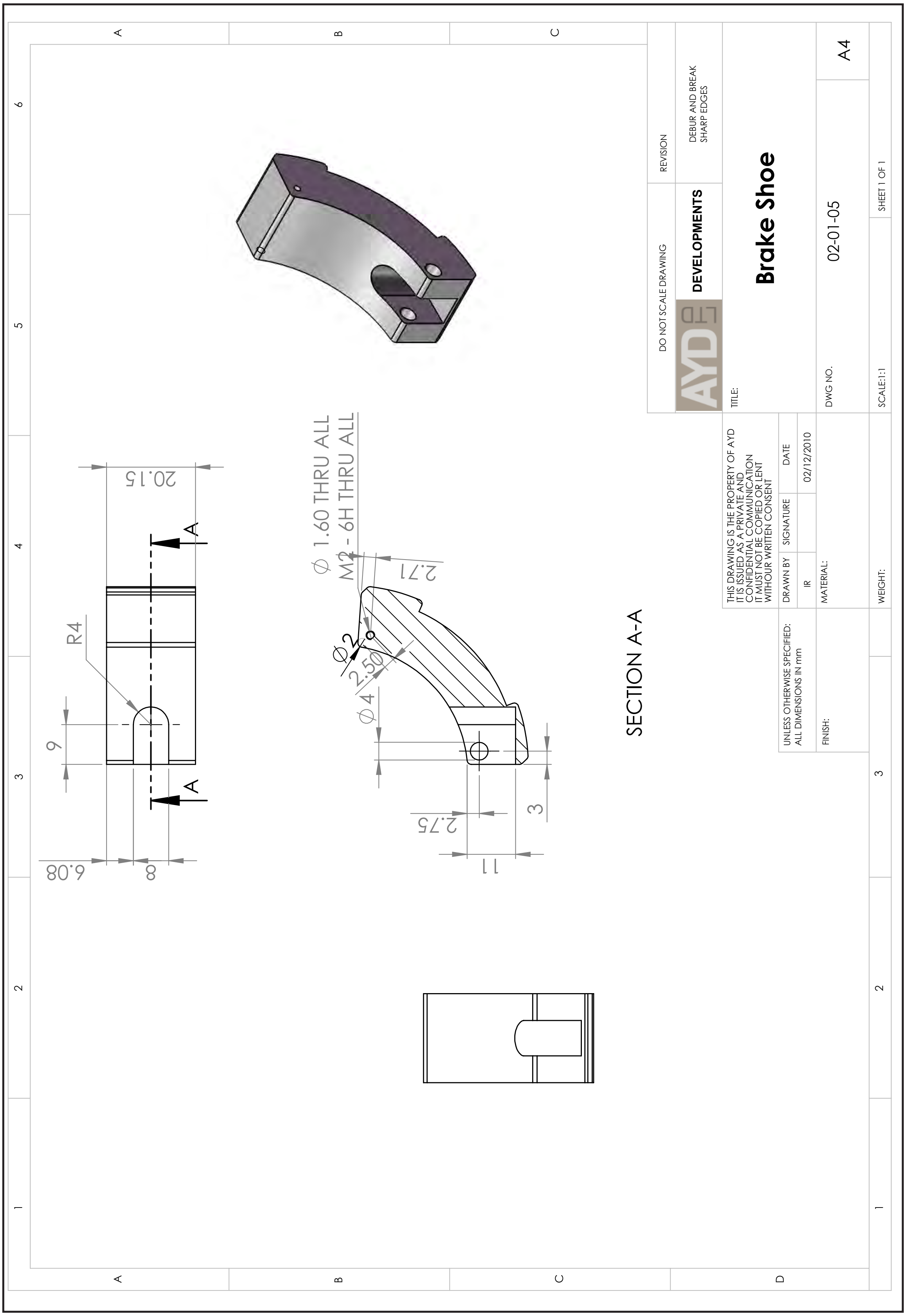


Figure A5.12 : Brake shoe

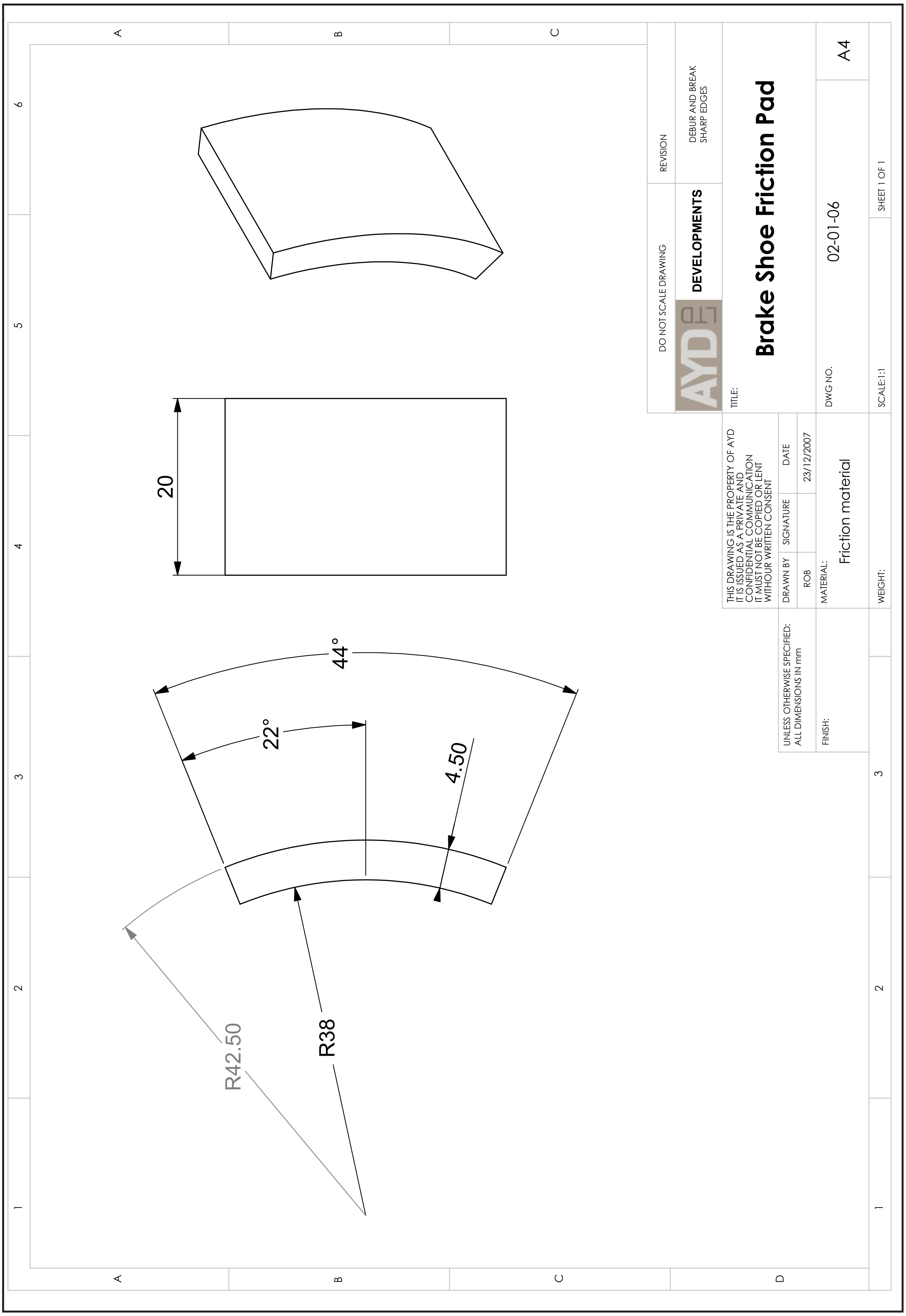


Figure A5.13 : Brake shoe friction pad

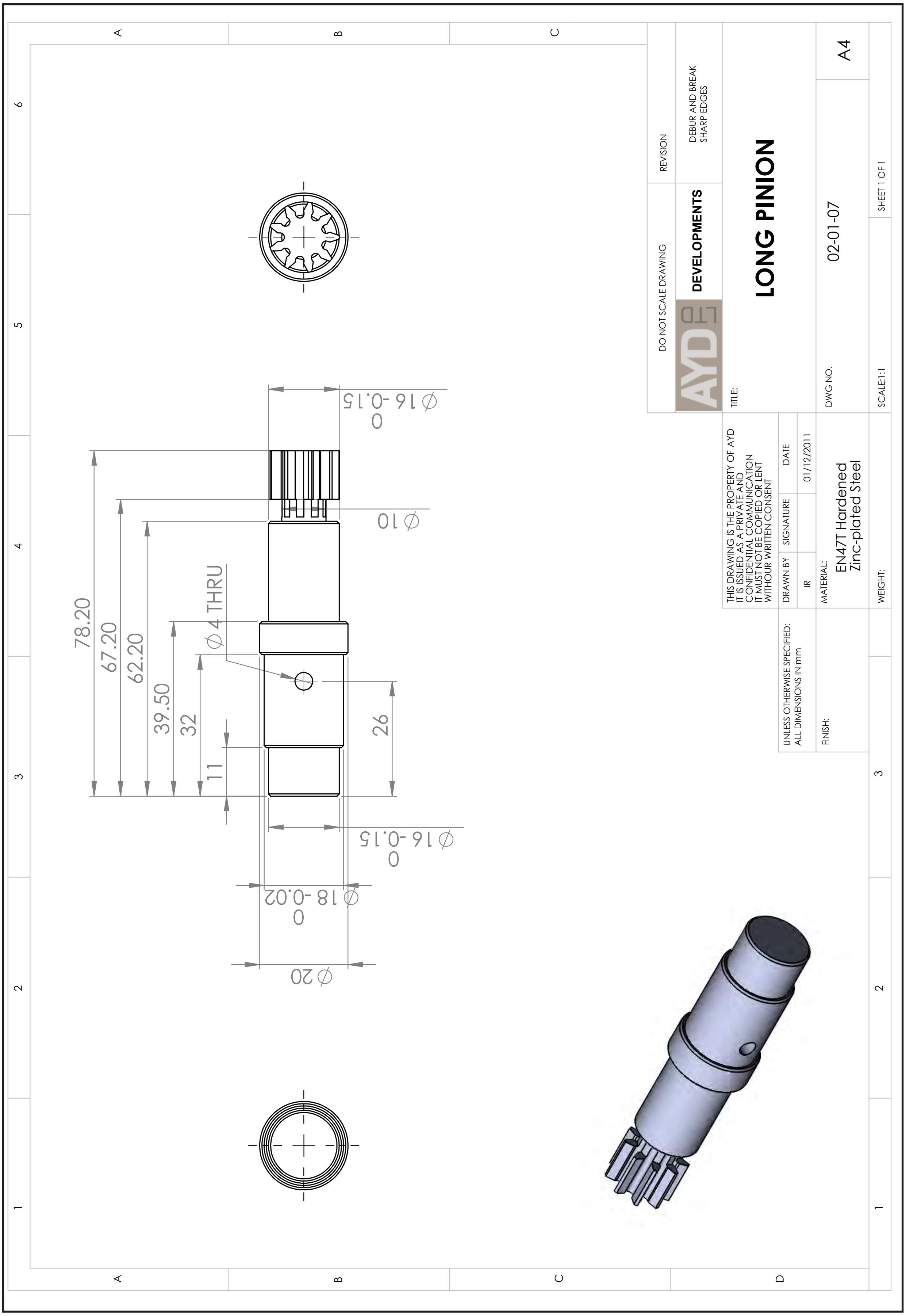


Figure A5.14 : Long Pinion

Appendix 6 - Correspondence - private

This has been removed